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HIGH-DUTY AIR-LIFT WATER PLANT

Discussion of the Air-Lift System Together with the Results of a Practical Test

BY C. F. IVINS

MOST FAILURES OF AIR-LIFT installations are resultant from the fundamental cause of lack of knowledge of local conditions, either applying air-lift to a condition that does not warrant it and that could be far more readily and cheaply met by the direct pumping system, or installing air-lift without a knowledge (1) of the supply—the yield and drop of the wells, (2) or the demand for a sustained or augmented consumption.

A fairly accurate knowledge of local conditions is obviously necessary to predict or estimate upon production and costs; but with this as an hypothesis, it is now possible to proceed in a scientific manner, recognizing the laws and principles that apply to this system and utilizing the data and information that have been compiled from experiments and observations conducted by manufacturers and bodies interested in this subject.

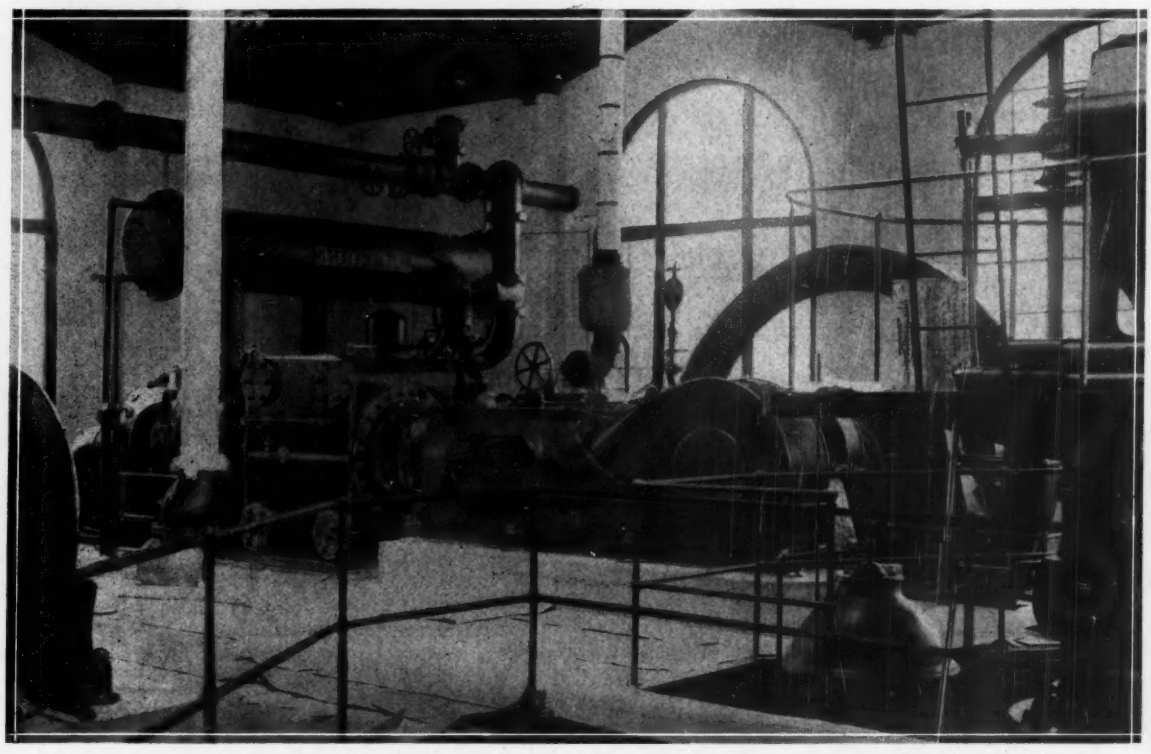


FIG. 1. INTERIOR OF POWER HOUSE AT PLAINFIELD WATERWORKS

Where the water supply to be drawn from is self-maintained, nonfluctuating and of known level, also

the lift and volume desired is specified, the results can be accurately and scientifically determined by adjusting the length of the air and discharge pipes to give the ideal submergence which will determine the air pressure in pounds required; viz.: Submergence in ft. x 0.434, while the selection of the size and ratio of air and discharge pipes will be determined by the amount of water desired, and which can be produced by the least amount of air.

Many municipal plants are working today giving economical and satisfactory results from the air-lift system, but few with an exact knowledge of their overall efficiency, or figures to show how far they fail to obtain the potential results. It is accordingly the purpose of this article to illustrate and describe a typical municipal water supply plant where the air-lift system has been in operation for one year, and the results of tests taken at the end of that time indicate that the system approaches the excellent figure of 40 per cent efficiency, and the fuel cost does not exceed $0.00557 per thousand gal.

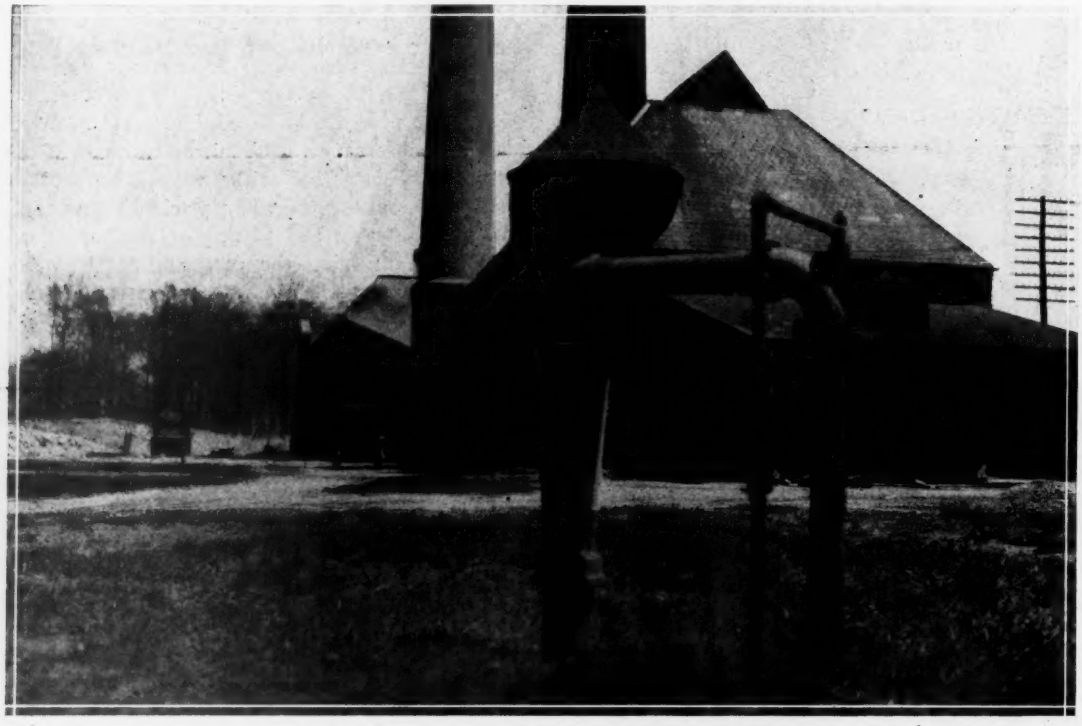


FIG. 2. DEEP GRAVEL WELL AT PLAINFIELD WATERWORKS

Character of Water Supply

THE plant in question, the Plainfield-Union Water Co., is situated in the vicinity of New York in a valley whose bed has a deep layer of sand and gravel above the rock, which lies at various depths below the surface to about 100 ft. The sand and gravel as well as the rock are most prolific in their water bearing qualities, and there is this peculiarity, that at this point there seem to be 2 separate and independent supplies of water. The gravel supply that stands at about 34 ft. below the surface of the ground and extends to the rock, saturating the sand and gravel and having a generally southern trend, is the product of the surrounding water shed and is quite susceptible to weather conditions; the levels of the wells fall with the continuance of a drought and rise commensurately with floods and wet weather. The flow of the wells is free, however, and the drop of the water in any of them responds actively when pumping on any in the vicinity.

The other source of supply is that which lies in the rock below the gravel strata, and is known to permeate the crevices and seams as far down as 400 ft., the depth to which any wells have yet been driven. This water is to a certain degree separate and distinct from the upper waters, and has certain chemical differences, which leads to the conclusion that the waters are sealed at the rock and have no connection, in that region at least, but probably take their source from some distant point where the deep rock strata outcrop, giving a connection between the surface and deep supplies which is well shown by the fact that when water has been reached in a deep well, it rushes up through the casing and stands at the same level as the water in the gravel wells in the vicinity, indicating a pressure on the rock water produced by a head equal to the height of the gravel water.

To exemplify this condition we have a deep well and a gravel well adjoining, and pumping from one does not affect the other, although wells of its own kind in the vicinity will be drawn down considerably.

This condition holds good regarding individual wells, but when the level in either set has been affected for a considerable period through continued pumping, then the level in the other set of well will begin to show its connection by a corresponding drop.

The Old Plant

ON the site in question was a plant of the ordinary direct suction type, equipped with 2 vertical duplex triple expansion condensing steam pumps of 5 and 6 million gallons per day capacity, that pumped the water from a series of wells, forced it into a 140-ft. stand pipe and maintained a working pressure of about 65 lb. The wells in this system were sunk through the gravel to the face of the rock, and fitted with a 20-ft. length of screen to admit the water freely and sustain the sides of the well; moreover, the casings were cut off at 20 ft. below the ground level and all joined in series by a suction pipe, sealing and connecting them to the pump cylinders which were themselves set in caissons sunk 20 ft. deep.

Plans were made to install an air-lift plant and it was desired to utilize the supply from the rock wells and differentiate the 2 systems, so that the suction method on the gravel could be used independently of the air on the rock, or they could be used in conjunction.

The Deep Wells

ABOUT 14 deep wells were drilled to a depth of between 350 and 400 ft., occupying an extended area of ground and separated by at least 20 yd. These wells, as well as all the other drilling on the premises, were drilled by P. H. and J. Conlin of Newark, N. J. They were 10 in. in diameter, having a casing driven down to and a short way into the rock, effectively sealing off the upper from the lower water. These wells were considered able to furnish more than an adequate supply for all emergencies, but should the event arise in which they might be drawn down excessively, it was determined to provide another source of supply and to tap the gravel waters and pump them by air.

By this arrangement they could alternate between the 2 sets, and as it were, give one a rest and allow it to assume its normal condition by drawing the supply from the other set, or else operate several wells of each set in conjunction, thereby maintaining a higher water level than would be affected by pumping from one kind alone.

To install an air and discharge pipe in one of the already existing gravel suction wells would appear to be the next step and have then an air-lift from the gravel, but here this difficulty presents itself. By so doing will we have a submergence that will produce satisfactory results, that is to say for our given lift, have we the right percentage of submergence? This is a simple matter of fact and information, and referring to tables we find that for a lift of about 70 ft., which we have roughly computed, the percentage of submergence will be between 60 and 65 per cent. To obtain this condition a 10-in. well was drilled and cased to the rock about 90 ft. deep, the size of the tools was then reduced to 8 in., and the well extended to a depth of 200 ft. below the surface. The well then stood 10 in. in diameter to the rock; from there it continued throughout its length 8 in. in diameter.

A pipe, plugged on one end, a trifle smaller than the well, fitted to a 20-ft. length of screen with a short reducing coupling was let down into the well, so that the reducing coupling rested on the shoulder of the rock where the well was reduced to 8 in., and suspended the pipe in the rock, while the screen had fitted to it a pipe that extended to the surface. The outlet pipe that cased the well in the first place was then withdrawn entirely, and we have here the result of the gravel water flowing in through the stream and down the pipe in the rock. It is, however, confined by the plug in the bottom of that pipe which effectually keeps the 2 waters separate, and itself serves as a well or repository for the gravel water, and provides the extra depth necessary for the specified submergence. Subsequent experiments proved this arrangement to be effective and satisfactory, and to warrant the drilling and equipment of several wells of this type. These wells, however, are intended to be auxiliary and to serve in time of emergencies or alternate with the deep wells in time of drought, and while they are considered independent of the rock wells, their level or static head remains about the same as that of the rock wells.

FIG. 3. EXHIBITION WELL WITH UMBRELLA TOP DEFLECTOR

First Test of Rock Wells

A TEMPORARY plant was installed upon the ground to test each well and determine as nearly as possible its yield, drop and character and if the result was not satisfactory, it was either abandoned altogether or continued down until better results were attained. In this way only could, with any accuracy, be determined the number of wells needed, the amount of water attainable, and the size and character necessary to produce the desired amount of water.

For this work a small Ingersoll-Rand steam driven air compressor was installed and pipes were laid to 3 wells, at the expense of the water company. The offers of 3 air-lift concerns to equip one well each in their own manner and run a competitive test, were then accepted, the company demanding that they should conform to standard form in conducting their

tests and compiling their results, and that the tests themselves should be conducted in the presence of and subject to the approval of the superintendent or his inspector. As these 3 wells were presumably all of about the same size, depth and capacity, a final standard of efficiency would be set for each concern, indicative of the duty of its respective type of plant, and conclusions of superiority would be drawn from that criterion.

This figure was the percentage of power theoretically used in raising from the well the actual number of gallons, the actual number of feet, to the amount of power consumed by the machine in producing the amount of air necessary for the work, or in other words, the percentage of water horsepower to air horsepower. This is assumed to be the efficiency of the system.

In Table I are shown the results of the test on well I, which was assigned to the Harris Air Pump Co. of Indianapolis, Ind., and the method used by all in keeping the results and comparing the records. This method of keeping the tests was found most inclusive and indicative of the performance of each well and can be applied separately, as well as collectively.

The final figure, 30.9 per cent efficiency, was the best obtained from any of the competitors and was considered very good, in view of the fact that the plant was a temporary and makeshift variety, and the compressor was not especially economical in its consumption of steam. The figures obtained from this preliminary test are valuable, however, in computing for future needs, for from this representative well, we know,

First—The number of gallons the well yields at maximum efficiency.

Second—Amount of air required.

Third—Drop, which, plus the static head gives the distance required to lift the water.

From these are derived the pressures necessary to start and run also the horsepowers and efficiency.

That almost any amount of water may be gotten from a well up to the limit of the yield of that well

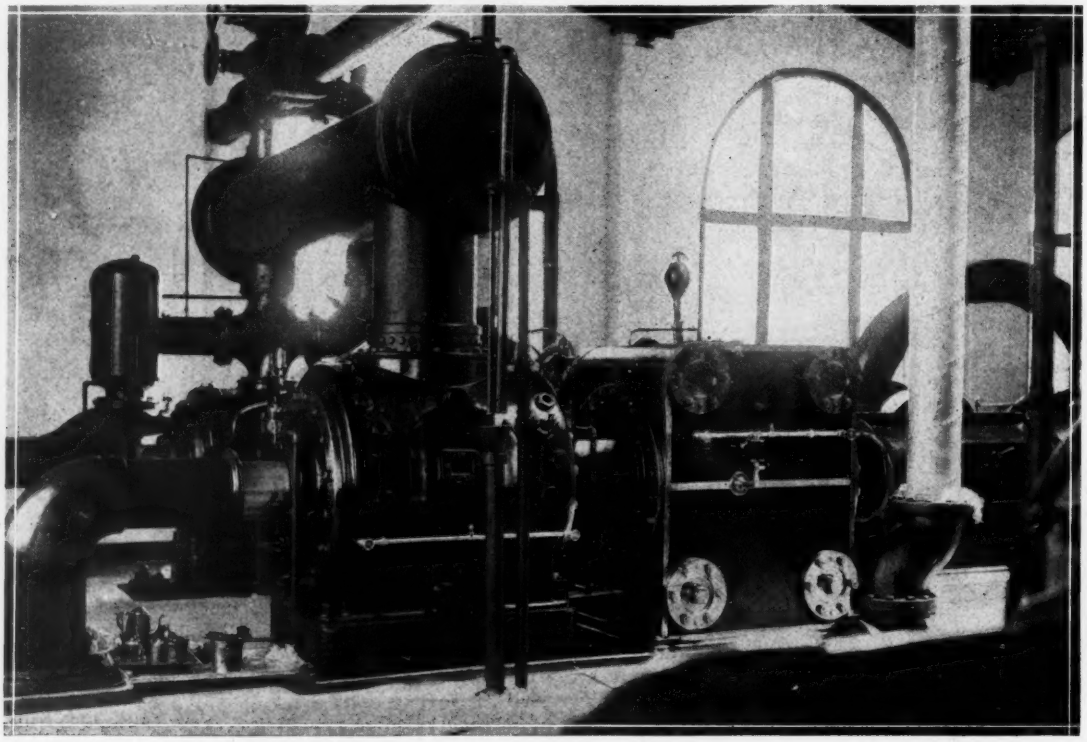


FIG. 4. AIR COMPRESSOR IN POWER HOUSE OF PLAINFIELD WATERWORKS

TABLE I. TEST WITH HARRIS SYSTEM DEEP WELL
No. 1, 300 Ft. Deep

Casing	93 ft. x 10 in.
Air pipe	158 ft. x 2 in.
Discharge	163 ft. x 6¼ in.
Static head	37 ft.
Pumping level	60 ft.
Drop	23 ft.
Elevation	5 ft.
Total lift	65 ft.
Submergence	98½ ft.
Percentage of submergence	60.2
Pressure start	55 lb.
Pressure running	47½ lb.
Gallons per minute	499
Water hp.	8.19
R.p.m.	92
Receiver pressure	51
Feet per minute	197
Air hp.	27.18
Air per gal	0.394 ft.
Over-all efficiency	30.9
Shut in press	42½
Temperature of water	54 Deg.
Duration of run	2 hours.

itself, is a reasonable statement to make for this system, for by forcing an unlimited supply of air through the pipe, we could practically blow the water out and draw it down to a point where the submergence would be so slight that we would be obtaining a minimum of economy and forcing our well to a point of least efficiency. What must be sought, however, is the relative point between air, water and lift that gives the greatest efficiency. Table II illustrates this and shows the result of test taken from one well to determine that point. With the compressor running at different speeds, the water production was measured and the pressures noted, and we were able to show the rise and fall in the relative efficiencies as the speed, viz.: air production was changed.

TABLE II. RELATION BETWEEN AIR, WATER AND PRESSURE

R.p.m.	120	100	80	75	65
Free air per minute	255	213	170	160	138 cu. ft.
Gallons water per minute	380	337	312	280	205
Drop of well	21 ft.	18.5 ft.	16 ft.	16 ft.	14 ft.
Air per gallon	672 ft.	633 ft.	545 ft.	572 ft.	673 ft.
Pressure	42	43	44	44	45
Efficiency	19	20.5	23.4	22.5	18.8%

It is to be noted that the greatest amount of water is produced by the greatest amount of air, and also the other extreme; but there is a point where 1 gal. is produced by the smallest fraction of a cubic foot of air, and this point corresponds with the point of greatest power efficiency, where the relation between air horsepower and the water horsepower is greatest, and as in this plant, so in any other plant can likewise be determined an input of air that will produce the most economical results.

The New Plant

PRELIMINARY test having proved conclusively that there was abundant water to be had, a large addition to the power house was built and without interfering with the working of the old plant, there were installed 2 Ingersoll-Rand cross-compound condensing air compressors, with Corliss steam gear and of the latest and most refined type OC. 3. The main receiver was connected to air mains of 4 in. and 6 in. diameter, and to these trunk lines, the wells were joined by 2-in. lines with provisions for cutting out any well or series of wells as desired.

The wells themselves were equipped by the Harris Air Pump Co. with its patented 20th Century foot piece and umbrella top discharge, and the discharge of the wells was carried by gravity to a 500,000-gal. reservoir, from which the steam pumps took their supply for distribution.

After this plant had been in operation about one year, it was decided to make a test and determine how the wells had been affected, and also what was the general working condition of the system. Under normal conditions of water production and steam consumption, 8 wells working simultaneously were giving an ample supply of water, and the compressor governed by the pressure regulator was making about 77 r.p.m. As it was obviously impractical to measure either the supply of air or the discharge of water from each well individually and thereby obtain the efficient working of each well, a much simpler and more ingenious method was employed in determining the theoretical percentage of submergence for the well in question and then proceeding to adjust the air throttle at the well, at the same time noting the change of pressure which would denote the change in submergence.

When the submergence was so adjusted that it comprised with the total discharge pipe the right percentage of submergence, we may assume that the well was working at its most economical point. The result of this adjustment is shown in Table III for each well and in each case an average was taken to indicate the general performance of the system. The length of discharge pipe in each well has been taken as 100 per cent, and knowing the lift to be somewhere between 80 and 90 ft., we find from tables that the percentage of submergency is between 55 and 60 per cent, consequently we admit just that amount of air to the well that will draw down the water to a point that gives the shut-in pressure that determines the correct submergency.

TABLE III. REPORT OF TEST MADE AT PLAINFIELD, N. J., MARCH 25th, 1912

No. of wells	3	4	5	6	8	9	10	11
Depth	350′	400′	350′	300′	304′	350′	350′	350′
Length of discharge (100%)	204	204	203	204	210	217	205	213
Starting pressure (lbs.)	66	60	60	65	65	65	65	65
Running pressure (lbs.)	59	55	55	57	56	60	54	55
Shut in pressure (lbs.)	53	51	50	54	51	54	48	50
Submergence, per cent	60	57.5	56.5	61	56	57	53.5	54
Submergence (ft.)	122	117	115	124	117	124	110	115
Lift (ft.)	82	87	88	80	93	93	95	98
Free air constant	.400	.432	.440	.387	.462	.447	.485	.489

Average submergence ..... 118 ft.
Average percentage of submerg..... 57%
Average lift ..... 89.5 ft.
Average constant ..... .443

The free air constant is an arbitrary figure, deduced from practice and theory, and denotes the fraction of a cubic foot of free air that should be required to lift one gallon of water through a given distance.

This simple adjustment resulted in a great saving of power for on returning to the power house, it was found that the compressor had slowed down to 67 r.p.m., the pressure being constantly maintained by the regulator, and that the wells were delivering the same amount of water. We can accordingly say that during the time that had elapsed since the system was first operated the wells had changed in some respects, and in other cases, the throttles were admitting more air than economy demanded. This is one instance where as much as 10 per cent of power may have been saved.

With these wells in proper adjustment, it was desired to determine the amount of water lifted, likewise the power required to produce the air for this work. As the water was flowing into the reservoir from a number of discharge pipes this could not be accomplished there, and therefore it must be measured as it passed through the steam pumps on its way to the stand-pipe with due allowance for fluctuations in the reservoir. The power required to compress the air was determined from indicator cards taken from the steam end of the compressor.

TABLE IV. TEST

Time, a.m.	10:05	10:20	10:35	10:50	11:05
Compressor, r.p.m.	67	67	67	67	65
Working pressure	69	69	69	69	68
Water lift reservoir	30⅝ in.	31⅝ in.	30 in.	33⅞ in.	32¼ in.
Rate of pump encounter	7743	8203	8654	9087	9561
Steam pressure	115	109	114	114	104
Average revolution of compressor per minute				66.6	
Average working pressure				68.8	
Revolution of pump in one hour				1818	
Size of duplex pump cylinders				20½ x 24-in. stroke	
Gallons per revolution				137.2	
Gallons pumped in one hour				249,429.6	
Gallons pumped per minute (displacement)				4157.1	
Estimate slippage, 10%				415	
Actual gallons pumped per minute				3742	
Size of reservoir				250 x 22 ft.	
Gallons in reservoir per inch				2487.5	
Loss in reservoir in one hour				1⅝ in.	
Loss in reservoir in gallons				5586.5	
Loss in reservoir per minute				93.1	
Actual gallons pumped per minute				3649	

In order to check the slippage of pump, a test was run on the reservoir with the wells alone, stopping the pump.

Time of test	15 min.
Water lift in reservoir increased	16 in.
Gallons pumped per minute	3666
Size of compressor used	Steam cylinder, 34 x 30 Air cylinder, 32¼ x 21¼ Stroke of all cylinders, 27 in.
Cu. ft. displacement in compressor per rev	23.77
Average revolution per minute	66.6
Cu. ft. free air per minute displacement	1583

3666 gallons per minute lifted 89.5 ft. equals 82.85 w.hp.
w.hp. divided by i.hp. equals 39.26% efficiency.

The results of this test are shown in Table IV, together with an incidental run with the pump shut down to check the slippage of the pump, and also check the actual pumpage.

From these figures we draw our general conclusions as to the economy and efficiency of the system. 3666 gal. p.m. lifted 89.5 ft. =

$$\frac{3666 \times 89.5 \times 8.33}{33{,}000} = 82.85$$

I.hp. in steam cylinder, 211.

$$\frac{\text{w.hp.}}{\text{i.hp.}} = \frac{82.85}{211} = 39.+ \text{ per cent}$$

What might appear at first to be a large discrepancy between the effective power and that actually expended is, however, readily accounted for when one considers the inherent characteristics of compressed air and realizes that there are certain unpreventable losses that must always be reckoned with; viz., mechanical friction, compression not isothermal, thermal change between receiver and point of application, friction of air in pipes, and slippage of water in the discharge pipe.

These losses can, however, be greatly reduced by proper provision, and in this case we may safely say that losses due to mechanical friction, and adiabiatic compression have been reduced to a minimum by providing a compressor of most refined and improved type, while the friction of the air in pipes, and slippage of water have been likewise affected by the design and selection of the well equipment. The losses, therefore, are those that we cannot expect to lessen substantially, due to the difference in temperature of the air at point of delivery and of application with the cold water.

Just what the fuel consumption of the air plant alone was, we were not able to determine, but taken in conjunction with the steam pumps which were forcing the same volume of water against a head of 150 ft., the entire plant was handling on an average day approximately 5,700,000 gal. and consuming 12 tons of coal.

Given $2.60 as a fair price per ton, this would be about $0.005 per thousand gallons pumped. The result of this installation may well be considered a conclusive and demonstrable denial to contentions and assertions that the air-lift as a system is extravagant and uneconomical; for, given its proper sphere of operation and due care and provision in selecting and installing, results may be realized that permanently recommend and establish its use.

COMBUSTION CHAMBERS FOR BOILERS

Slight Changes Which Have Raised Efficiency

By Everard Brown

NOT a great many years ago the proper size of the combustion chamber, especially as regards height, under the horizontal, water-tube type of boiler, was not given much consideration. In fact, the tendency was to keep the boiler as low as possible in order to keep down the initial cost of installation. More modern practice, however, has demonstrated the fallacy of this idea and has proved that, up to a certain size, of course, the larger the combustion chamber the better will be the operating results, both economic and otherwise.

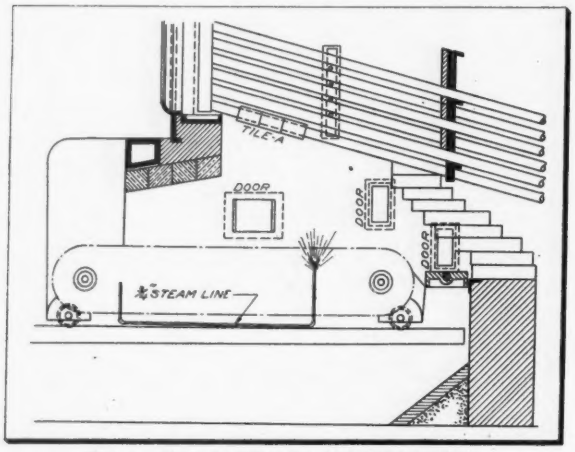


FIG. 1. TILE BAFFLE TO AID COMBUSTION

Unfortunately, there are many of the old style boiler plants still in operation and it is up to the operator in each case to do the best he can with the equipment at hand. He is generally expected to make at least a reasonably good showing in comparison with the more modern plant and often has the central station figures to combat. In addition to this there is also the smoke problem to contend with if the plant is located in a city where such an ordinance exists and is enforced.

The question of what to do under such conditions is not always an easy one. The boiler plant may be too good to justify dismantling it and building a new one and to raise the boilers so as to get a larger combustion chamber involves considerable expense; besides, this is not always possible because of the limiting height of the boiler house itself. On the other hand, if no change is made there is the continued high cost of steam and possibly at times a shortage as well, and a large portion of the value of the coal going out of the stack in the form of smoke.

Under such conditions the owner of the plant is most likely to ask the man in charge as to the best thing to do and to give him an opportunity to solve the problem, if possible, before calling in outside aid. Advantage should be taken of such an opportunity—

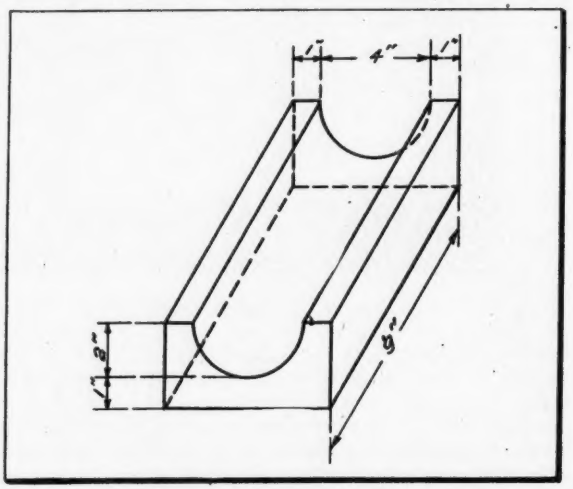


FIG. 2. FORM OF TILE USED ON WATER TUBES

in fact, the wise operator does not wait for its coming but has his plant at the top notch all the time—and often a little investigating or experimenting will disclose a method or device whereby conditions can be materially improved.

As an example of this, Fig. 1 shows a rather low combustion chamber under a horizontal, water-tube boiler that was considerably improved by the appli-

cation of 3 9-in. tile on the lower row of tubes the full width of the boiler. This tile is shown in Fig. 2. There are other instances as well where better results were obtained by similar applications.

Under the boiler shown in Fig. 1 a rather long flame coal is burned, consequently there is not sufficient space for the complete combustion of the slow burning gases before they strike the comparatively cool surfaces of the lower tubes. This impinging against the colder surfaces naturally lowers the temperature of the gases below their burning point so they pass on through the boiler and out of the stack unconsumed, resulting, of course, in a direct loss of a portion of the heating value of the fuel.

At a point where it was thought most of the gases and flame impinged, the tubes were covered with tile as shown, leaving, however, a space of about 6 in. between the ends of the first tile and the inside face of the arch. This space is intended to prevent the front ends of the tubes from becoming dead, as it were, for if the tile are brought up close to the arch there will no doubt be considerable space above them that will not be reached by the hot gases. Of course, this arrangement does not prevent all of the flame or unconsumed gases from striking the tubes; but, nevertheless, the conditions have been much improved both as to getting a better evaporation per pound of coal burned and a less amount of smoke from the stack.

There are 2 difficulties that might be mentioned in connection with the above scheme; one is the trouble and expense in maintaining the tile and the other is the accumulation of dust, etc., on top of the tile. As regards the former, it has been found that ordinary first grade fire brick tile will not last much over 2 or 3 months on account of breaking and for this reason it would probably be better to put in extra hard burned or possibly steam pressed tile which might give sufficient density to avoid this trouble. As for the accumulation of dust, etc., on top of the tile, this would, of course, be natural; but if provision was made to blow it off periodically, this should not be a serious difficulty.

Another arrangement that has given equally good, if not better results than the tile is shown in Fig. 3. It will be noticed that the arch has been extended somewhat, approximating to a degree the Dutch oven idea. In addition to this, air is admitted through an inlet B in the boiler setting and is carried along in the direction shown by the arrows until it reaches the fire and is in a preheated state. The quantity of this air passing over can, of course, be governed. A plate is also provided for the purpose of cutting off part of the cold air current at the hottest part of the fire. The steam jet shown in both sketches is, of course, rather common practice and assists in preventing the formation of clinker.

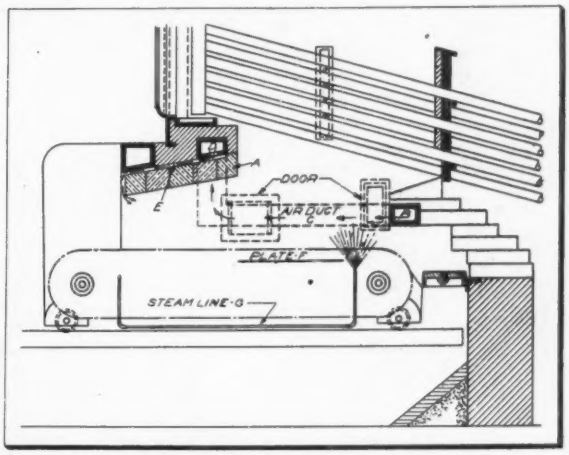


FIG. 3. AIR HEATING SCHEME, BAFFLE AND STEAM JET INSTALLED TO OVERCOME DEFECTS OF POOR SETTING

Three particular features of this scheme, the Dutch oven effect, even if only slight; the preheating of the air before it reaches the fire and the shutting off of the cold air from underneath the hottest part of the grate are improvements which both in themselves individually or collectively should prove of considerable benefit, especially in installations similar to the two shown. Furthermore, the cost of trying it out should not be much in any case, consequently if it does no good there is no particular harm done.

There are "kinks" in generating steam, as well as in any other business and those shown are only two of many others. They are given here as merely illustrative of what a little thought and experimenting will do.

HEATING AND VENTILATING LARGE BUILDINGS

Methods of Ventilating Office Buildings and Separate Rooms

By Charles L. Hubbard

IN treating of the subject of ventilation in the present series of articles we shall confine ourselves principally to the conditions found in office buildings and, especially to the rooms or suites of rooms where it is desired to improve the air supply without too great an expense, or without making extensive changes in the construction of the building itself. The design of a ventilating system for a public building is entirely beyond the scope of the present article and would be of but little value to the average engineer in charge of a power and heating plant. Some general data will be given in regard to the size of ducts, air velocities, etc., and also a brief outline of the best arrangements for air distribution in office buildings.

Beyond this, our attention will be given chiefly to the ventilation of isolated rooms.

Introduction of Fresh Air

ONE of the simplest methods of introducing air on a fairly generous scale to first floor rooms, such as stores and banking rooms, is to place indirect stacks beneath the floor and admit fresh air from outside under a gravity flow. Methods have already been given in a previous article for determining the amount of heating surface in cases of this kind.

The general arrangement of a heating stack, casing, and warm-air connection to the room above, is shown in Fig. 1. In this particular case the heater is located in a cold-air room and the bottom of the casing is left open for the admission of cool air. If the stack is in a used room, the bottom of the casing should be closed in and the space beneath the heater

connected with the outside air by a duct of suitable size. A common method is to allow 2 sq. in. area in the connecting air ducts for each square foot of heating surface in the stack. The mixing damper is for regulating the temperature of the entering air without varying the quantity to any extent.

As will be seen from the cut, the relative proportions of hot and cold air may be varied by changing the position of the mixing damper. When it is horizontal, all hot air will be delivered to the room, and when set at the other extreme, the hot air will be

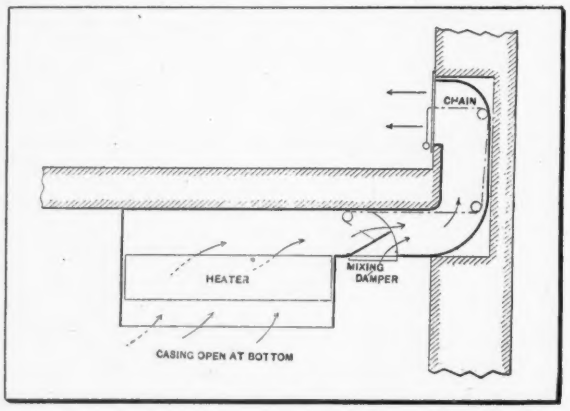


FIG. 1. SIMPLE METHOD OF INTRODUCING AIR

cut off and all cold air will be admitted without changing the sectional area of the duct. The air spaces above and below a heater of ordinary size, say, up to 150 sq. ft. of surface, should not be less than 10 in., and 12 is better if there is sufficient head room. When the inlet register is located at an elevation above the heads of the occupants care should be taken to arrange the mixing damper so that the cool air will pass up the back of the flue and be delivered into the room at the top of the inlet register. Otherwise the hot and cool currents of air will not mix properly and there will be a down draft of cold air upon the heads of those near the register.

In Fig. 2 the cool air is represented by the straight arrows and the warm air by the crooked ones, and the manner in which the 2 currents mix when properly introduced is clearly indicated in the cut.

Ventilation in Office Buildings

MENTION has already been made of the general method of distributing the air in large office buildings where trunk lines are formed by furring down the corridor ceilings as shown in Fig. 3A. The main supply flues from the fans are carried up through the building at points adjacent to the corridors and openings made with the distributing ducts at each story.

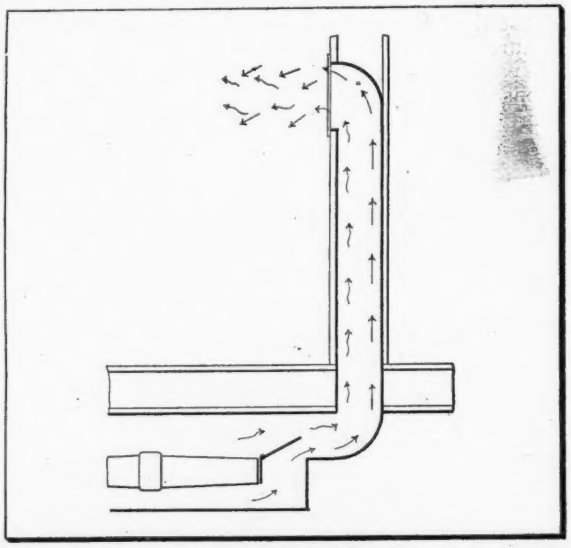


FIG. 2. ARRANGEMENT TO DELIVER THE COLD AIR AT THE TOP OF REGISTER

The velocity in the main uptakes may be from 1200 to 1500 ft. per minute, if suitable deflectors are placed over each outlet to turn the proper amount of air into the different ducts so as to secure a proper distribution throughout the building. The velocity in the horizontal ducts may be taken at about 1000 ft. per minute. A uniform air flow into the rooms through the registers is commonly obtained by placing one or more layers of perforated metal back of the register, as may be required to give the required velocity. This arrangement is for the usual case where the building is heated by direct radiation and the air is supplied to all rooms at a uniform temperature of 68 to 70 deg.

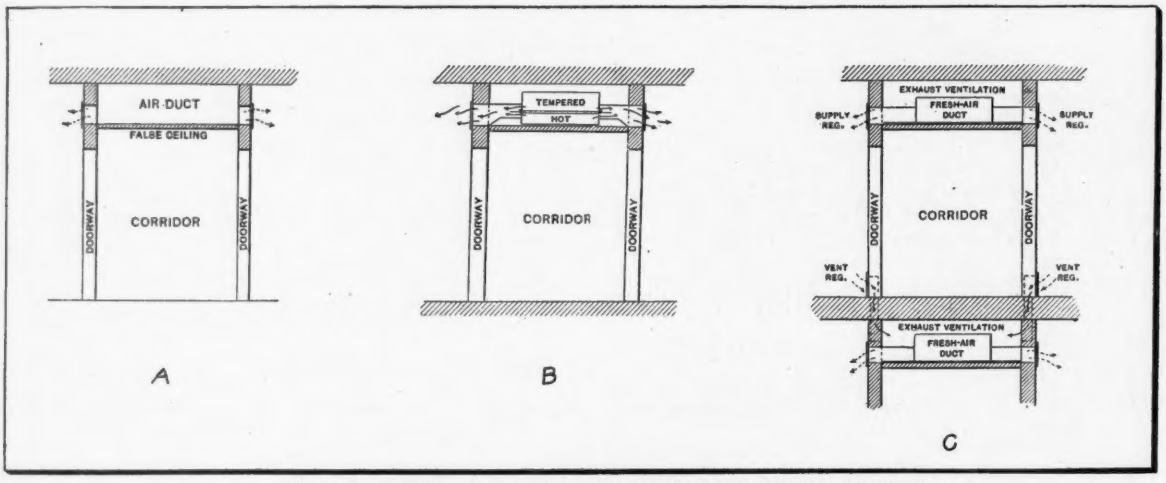
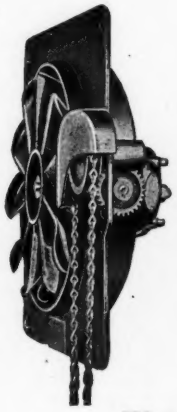


FIG. 3. METHODS OF VENTILATING EMPLOYED IN OFFICE BUILDING

Sometimes a portion of the rooms are heated as well as ventilated by warm air. In this case, in order to provide a means of temperature regulation without reducing the air supply, 2 ducts are carried in the furred space, one carrying about 3/5 the air supply at a temperature of 68 to 70 deg., and another duct



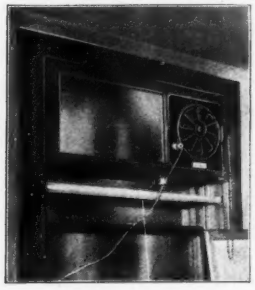


FIG. 4. VENTILATING FAN
FIG. 5. METHOD OF USING VENTILATING FAN

carrying the remainder at a temperature of, say, 100 to 120 deg. according to the outside temperature. Mixing dampers are provided where the branches leave the ducts so that a mixture may be delivered to the rooms of the proper proportions to give the desired temperature. In heating the air for this system, the full volume is passed through a primary heater which raises its temperature to about 68 deg., and then 2/5 of that is passed through a secondary heater which raises the temperature of that portion to the higher figure mentioned above.

FIG. 6. PORTABLE HEATING AND VENTILATING OUTFIT

A more complete system of ventilation is illustrated in Fig. 3C. In this case the rooms are heated by direct radiation as in the first case described, and illustrated in Fig. 3A, the furred-in space around the fresh-air duct is utilized for exhaust ventilation, being connected with a special fan for this purpose. The method of connecting the vent registers with this space is shown on the lower floor in the cut, and is carried out in the same manner for all floors below. The velocity through registers should be limited to 400 ft. per minute for the over-all area, else there are likely to be unpleasant drafts unless special diffusers are provided.

Ventilation of Individual Rooms

AS previously stated, ventilation in all of the rooms of an office building is the exception rather than the rule, and those in charge are often called upon by tenants who are more particular than others, or better, who more fully appreciate the value of fresh air, to provide some simple means for ventilating isolated rooms. There are several devices for this purpose upon the market, two of which are shown in Figs. 4 and 6.

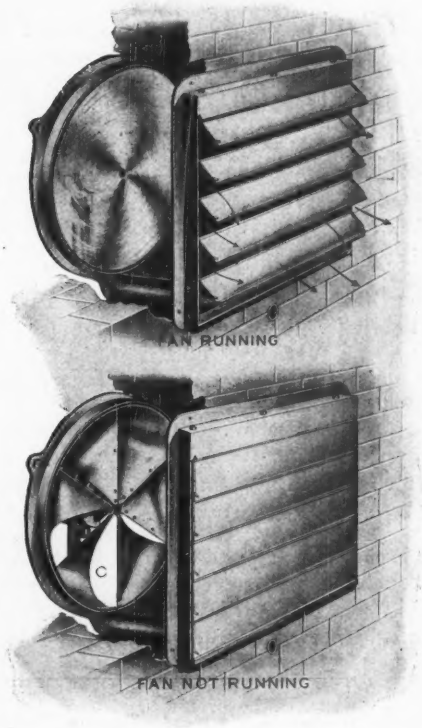


FIG. 7. DISCHARGE FAN AND SHUTTER

The first of these consists of a small disk fan with direct-connected motor, set in a metal frame. This is provided with a gear and chain wheel by means of which the fan may be made to discharge at any angle desired or may be entirely reversed.

One of the best arrangements for a device of this kind is to set it to discharge under the raised lower sash of a window, thus drawing from the bottom of the room instead of the top. By lowering the upper sashes slightly, cool fresh air will be drawn in to take the place of that which has been forced out at the bottom, and if not too great in amount, will mix with the hot air at the upper part of the room without sensible drafts. By the reverse attachment, the fan may be made to draw in fresh air if desired, and if set to throw it upward on an angle, it may often be made to give good results in this way. When air is drawn into a room by the fan, the vent is usually through a transom into the corridor.

Fans of this kind are usually from 12 to 15 in. in diameter and the driving motor may be supplied from an ordinary light socket. They will furnish good ventilation for 5 or 6 people without any special apparatus except the casing in the window or transom. This particular fan is made by the American Blower Company, of Detroit, Mich.

The difficulty of bringing in any considerable quantity of air with the arrangement just described is due to the fact that it provides no means for warming the air. The direct radiator in a room is proportioned to warm it in zero weather, so that for most of the heating season there is a surplus of heat which may be utilized for warming a certain amount of outside air for ventilating purposes, provided it can be mixed with the hot air at the top of the room without producing unpleasant drafts.

A portable apparatus, provided with a heater for warming the air, and having a capacity of approximately 15000 cu. ft. per minute, is shown in Fig. 6. The air is delivered to the machine at the bottom through an opening cut in the iron base, and may be brought from the most convenient point, which may be through a panel at the top or bottom of a window, or by means of a special opening in the wall or floor. The fan is of the squirrel cage type, driven by a direct-connected electric motor. The heater is made of pressed steel to give it lightness, and is connected with the regular piping system of the building. The opening through which the warm air enters the room is in the top of the casing and is provided with a fine wire screen for removing the coarser dust contained in the air.

This apparatus is made by the Garner Ventilating Co., New York.

A convenient discharge fan and shutter is shown in Fig. 7. This is adapted to the ventilation of toilets, smoking rooms, etc., where air should be drawn from the space to be ventilated rather than forced in under pressure. The fan and motor are supported from the shutter frame, which is of iron, and may be set in a wall opening as indicated. This outfit is made by the I L G Electric Ventilating Co., Chicago.

An application of the Sturtevant Ready-to-Run ventilating set is to place it over the closets in a small toilet, and arrange it to draw the air over the fixtures and discharge it outbound.

This piece of apparatus consists of a small centrifugal fan and direct-connected electric motor and is adapted to a variety of uses in the ventilation of small isolated rooms. This is made by the B. F. Sturtevant Co., Hyde Park, Mass.

Ready-made apparatus has been shown, as it is more convenient to install when only a room or two are to be provided for. If desired, good practical ventilating outfits may be gotten up under the direction of the engineer along the lines described, by using standard equipment and calling in a good carpenter and sheet metal worker.

In determining the volume to be moved in any case, provide for a supply of 30 to 50 cu. ft. per occupant per minute, according to local conditions, bearing in mind that 15 or 20 are much better than nothing. When toilets and smoking rooms are ventilated by a discharge fan, provide for a complete change of air in the room every 10 or 12 min. if possible, and more when conditions are particularly bad.

STANDARD BOILER SPECIFICATIONS

TOWARD the desired goal of uniformity in boiler practice, the National Association of Tubular Boiler Manufacturers has taken a long step. Standard specifications for materials, design and dimensions of horizontal return tubular boilers have been adopted by 34 leading firms, which will handle boilers on these specifications as standard. Other specifications will be made special, but necessarily at a greater cost and longer delivery as is always the case with special goods. The firms are as follows: Bass Foundry & Machine Co., Fort Wayne, Ind.; John Brennan & Co., Detroit, Mich.; Broderick Co., Muncie, Ind.; Wm. Bros Boiler & Manufacturing Co., Minneapolis, Minn.; The Brownell Co., Dayton, Ohio; The Casey-Hedges Co., Chattanooga, Tenn.; Chandler & Taylor Co., Indianapolis, Ind.; R. D. Cole Manufacturing Co., Newnan, Ga.; Erie City Iron Works, Erie, Pa.; S. Freeman & Sons Manufacturing Co., Racine, Wis.; Frost Manufacturing Co., Galesburg, Ill.; Gem City Boiler Co., Dayton, Ohio; Houston, Stanwood & Gamble Co., Cincinnati, Ohio; E. Keeler Co. Williamsport, Pa.; Kewanee Boiler Co., Kewanee, Ill.; Lombard Iron Works & Supply Co. Augusta, Ga.; Lookout Boiler & Manufacturing Co., Chattanooga, Tenn.; McIlvain & Spiegel Boiler & Tank Co., Cincinnati, Ohio; Milwaukee Boiler Co., Milwaukee, Wis.; John H. Murphy Iron Works, New Orleans, La.; Murray Iron Works Co., Burlington, Iowa; John O'Brien Boiler Works Co., St. Louis, Mo.; Phoenix Iron Works Co., Meadville, Pa.; John Rohan & Son Boiler Works Co., St. Louis, Mo.; J. S. Schofield's Sons Co., Macon, Ga.; Dan Shea Boiler Works, Memphis, Tenn.; Southern Engine & Boiler Works, Jackson, Tenn.; Topeka Steam Boiler Works, Topeka, Kas.; Tudor Boiler Manufacturing Co., Cincinnati, Ohio; Union Iron Works, Erie, Pa.; Henry Vogt Machine Co., Louisville, Ky.; Walsh & Weidner Boiler Co., Chattanooga, Tenn.; C. J. Walton & Son, Louisville, Ky.; Joseph F. Wangler Boiler & Sheet Iron Works Co., St. Louis, Mo.

The standard specifications are quoted below.

The dimensions and rating of standard stationary tubular boilers, together with the sizes of the parts and usual fittings, will be found tabulated in the Practical Engineer Data Sheets. Boilers to conform to these specifications shall be made and equipped as there indicated and as described herein. All boilers shall be of such strength that the factor of safety shall be not less than 5.

Materials

THE shell, heads and covering strips of the standard boiler shall be of flange steel as described in the standard specifications of the Association of American Steel Manufacturers.* All such plates shall be marked 60,000 lb. tensile strength and in building the boiler the plates shall be so placed that these stamps are plainly visible on the outside. These plates shall have the full specified thickness at the edges and shall meet the tensile, quenching and bend tests described.

The tubes shall be standard quality lap-welded mild steel of standard manufacture. The thickness

*Specifications for flange steel are: Tensile strength, pounds per square inch, 55,000 to 65,000; yield point minimum pounds per square inch, 0.5 tens. str.; elongation in 8 in., minimum, per cent, 1,450,000 ÷ tens. str.; manganese, per cent 0.3 to 0.6; phosphorus, max. per cent, basic 0.04, acid 0.05; sulphur, max. per cent 0.05.

of the metal shall be 12 b.w.g. for 3-in. tubes, 11 b.w.g. for 3½-in. tubes and 10 b.w.g. for 4-in. tubes.

The rivets shall be of boiler rivet steel as described in the Standard Specifications of the Association of American Steel Manufacturers† and of proper size to suit the size of the hole and the thickness of the plates and to form up heads equal in strength to the pressed heads of same.

The diagonal braces shall be weldless of the same quality as the flange steel previously described and pressed from the solid plate, or else forged from steel bars. The number of braces to be used shall be computed on an allowance of not more than 7500 lb. of load per sq. in. of section of brace neglecting, in this, the inherent strength of the heads and the slight angularity of the braces.

Suitable through rods and braces shall be installed below the tubes when necessary to sustain the pressure. These shall be of steel and shall be computed with the same allowance as diagonal braces.

Braces as above described shall be carefully placed so that the pressure on the flat surfaces, both above and below the tubes, shall be as nearly equally distributed as possible.

For any boiler head the area to be supported by braces shall be the surface included within lines drawn 2 in. from the outside of the tubes and 3 in. from the inside of the shell.

Design

THE longitudinal seam of the shell shall be of the butt joint type having inside and outside covering strips and either double, triple or quadruple riveted as indicated in the Data Sheets.‡

The circumferential seams shall be of the lap type of joint and single riveted.

In all joints the size, number and spacing of the rivets shall be such as to provide the strength necessary to maintain the factor of safety of 5.

The tubes shall be placed in vertical and horizontal rows with ample space between the adjacent tubes and also between tubes and shell for the circulation of water and also with a large steam space above the water line.

Boilers 44 in. or less in diameter shall have a manhole above the tubes and a handhole below the tubes. These may both be in the front head or the handhole may be in the front head and the manhole in the back head. Boilers 48 in. in diameter shall have 2 manholes, both in the heads, one above the tubes and one below the tubes. They may both be in the front head or one manhole may be in the front head below the tubes and the other in the back head above the tubes. Boilers larger than 48 in. shall have 2 manholes, one of which shall be in the front end below the tubes, the other shall be above the tubes in either head or preferably may be placed in the shell.

†Specifications for boiler rivet steel are: Tensile strength, pounds per square inch, 45,000 to 55,000; yield point minimum pounds per square in, 0.5 tens. str.; elongation in 8 in., minimum, per cent, 1,450,000 ÷ tens. str.; manganese, per cent 0.3 to 0.5; phosphorus max. per cent, basic 0.04, acid 0.04; sulphur max. per cent, 0.045.

‡The butt type of joint is specified because the lap joint has been proved dangerous when used for the longitudinal seams. Defects in the lap seam usually develop where the parts are hidden. Inspection cannot reveal them. Domes are not specified because they are mechanically bad and are not necessary. The fixed, or "plug hat" type of dome is awkward at best, it tends to weaken the boiler and frequently makes the use of the lap type of joint necessary on the longitudinal seam.

The independent type of dome can be used on any boiler, but it will probably be found to be better engineering to apply it in the form of a steam separator near the engine throttle instead of as a dome near the boiler.

The manufacturers who have adopted these specifications do not recommend domes or lap joints when used on longitudinal seams.

The opening in the manhole shall not be less than 10 by 15 in. Each manhole shall be equipped with a heavy plate, bale, bolt, nut and gasket.

Manholes placed in the shell shall be properly reinforced with a suitable steel saddle.

All openings 2 in. in diameter or larger shall have suitable steel reinforcing flanges riveted on. Cast-iron flanges or nozzles shall not be used.

Boilers for 125 or 150 lb. working pressure shall have the feed opening in the front head over the tubes and it shall be provided with a brass bushing and an internal feed pipe extending from the front end of the boiler to within about 3 ft. of the rear head, thence across to the side of the shell terminating in an elbow for discharge below the top row of tubes.

Boilers for 100 lb. working pressure shall be arranged to feed through the blowoff connection and the internal feed pipe be omitted.

Each boiler shall be provided with 4 steel lugs or brackets, 2 on each side for supporting the boiler, except that boilers 78 in. in diameter or boilers 20 ft. long shall be provided with 8 such lugs, 4 on each side of the boiler and all of them arranged in pairs.

Workmanship

ALL rivet holes, and all tube holes shall be either drilled from the solid or punched small and reamed to size. After drilling or reaming the plates shall be taken apart and all burrs removed and the tube holes chamfered with a rose reamer.

The edges of the plates shall be beveled to an accurate calking edge and after being riveted shall be calked tight with a round nose tool.

The tubes shall be carefully placed, expanded tight in the heads with a roller expander and the ends carefully beaded over against the head at both ends.

Water Column

THE water column shall be provided with 3 gage cocks and shall be placed so that the lowest gage cock shall be 2 in. above the tops of the uppermost row of tubes. The middle gage cocks shall be 3 in. and the uppermost gage cock 6 in. above the lowest gage cock measured vertically.

Tests

ALL material used in the construction of the boiler shall be rigidly tested as called for in the above specifications and the completed boiler shall be subjected to a steady water pressure 50 per cent above its proposed working pressure and made tight.

When specially requested the boiler manufacturer will furnish copies of the record of tests of the plates as made by the steel manufacturers. These reports shall be deemed sufficient indication of the quality of steel used.

Equipment

A COMPLETE standard boiler equipment includes: The bare boiler with lugs; cast-iron front with anchors; grate bars with bearers; rear arch bars; rear ash pit door and frame; safety valve with nipple to connect it to boiler; steam gage with siphon; water column with gage glass, 3 gage cocks and pipe connections to boiler; blowoff valve; check valve; stop valve for water feed line; smokestack and guys. (Stacks less than 60 ft. long shall have 4 guy eyes, and guys 6 times as long as the stacks. Stacks 60 ft. long or longer have 6 guy eyes, 2 sets of 3, and guys 10 times as long as the stack.) A stack plate or nozzle shall be included with each full front boiler.

Electrical Machinery and Appliances

For Generating, Transmitting and Using Current

SPARKING AND COMMUTATION*

Commutator Defects and Setting of Brushes

By William C. Thorne

HIGH bar may be due to working loose or being harder than others. Tap lightly with a hammer and see if solid; if so, cut down with a file level with the others.

Low bar may be softer than the other or may have been hit. The only remedy is to turn the others down to the level of the low one. A high or low bar is detected by a click at each brush every revolution or may be detected by letting the finger rest on the brush

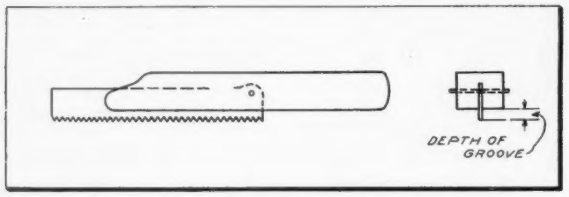


FIG. 4. TOOL FOR UNDERCUTTING MICA

tension spring, while the high bar goes under the brush. In case a high bar is loose a repair job must be done at the shop.

Hard Mica and Undercutting

OFTEN a machine has mica which is too hard and will not wear down as fast as the copper. This causes the brushes to break contact and spark, this spark being of a damaging nature. The remedy is to cut down the mica to not more than 1/16 in. below the surface of the bars; 1/32 in. is better. A tool for hand use is easily made by grinding a hack saw blade so that it is the same thickness as the mica. Fix it in a block of wood as shown in Fig. 4. The depth of cut in the handle regulates the depth of cut in the mica. A narrow chisel can also be used. Don't have the slot too wide as it will cause the brush to drop and chatter at each cut. After cutting smooth off the edges and polish. It is well to consult the maker before resorting to undercutting the mica to find if it is all right for this particular type of machine. A different brush may be needed on account of the changed nature of the surface. More care will be needed that the space between the bars does not fill up with oil, dirt, copper dust and carbon. Most of the dirt that does collect will be thrown out by the centrifugal force, but oil, copper chips, etc., may stick as in a paste.

*Continued from page 940, Oct. 1 issue.

Brushes

BRUSHES must be of ample size to carry the current and not too hard. Copper brushes are generally used for low voltage and heavy current such as plating and also some styles of lighting machines. Carbon brushes are generally used for higher voltage power and lighting service. Carbon brushes are generally set tangential to the commutator as shown in Fig. 5. Carbon brushes are generally self lubricating. A hard carbon brush is used in railway work and a softer grade for stationary work. Carbon brushes are used for variable loads and also where the machine is to be reversed at times, as they are set tangential to the commutator.

Metallic brushes offer too easy a path to the current leaving the bars going from under the brush and allow the current to arc; but carbon brushes have enough resistance to force the current through the other coil to the brush in spite of self induction. Carbon brushes on account of their high resistance are especially valuable on varying loads as they do not allow so much sparking and therefore do not require so much shifting. In buying brushes it is well to write either to the maker of the machine or to a manufacturer of brushes and explain the conditions under which the brushes will work, such as the peripheral speed of the commutator, width of bars, brush tension, load, number of brushes, amperage, voltage, and temperature of the room and get their advice. Brushes are hard, soft and medium, and self lubricating carbon, graphite, plumbago and special compositions for different conditions of service.

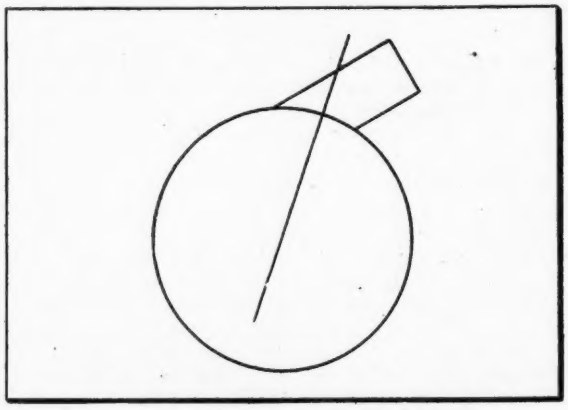


FIG. 5. TANGENTIAL POSITION OF BRUSH

As graphite or plumbago is a natural lubricant, no lubricant is needed with these brushes. They are especially valuable with soft copper commutators, and in hot rooms where lubrication is not satisfactory. Make a trial of different ones until a satisfactory one is found. All brushes should be copper-plated to help them carry the current, but the copper must be kept trimmed back from the bearing face for ⅛ to ¼ in. so it will not touch the commutator. Copper plated brushes will probably last longer. The area of the contact surface of the brush is of course determined by the designer and the operator does not need to worry about that. In general, it is better to buy self lubricating brushes than to use commutator compounds. Care must be used in using compounds to get satisfactory results. Hard brushes may be lubricated by lightly drawing a piece of paraffin wax over the commutator or by slightly moistening a rag with vaseline and applying the same way. A little kerosene is often beneficial with graphite brushes. In a general way many commutator compounds are not entirely satisfactory, as even when lightly used a film of wax is formed on the surface of the commutator.

Fluctuating voltage is often caused by foreign matter on the commutator. Light mineral machine oil very sparingly applied is a good lubricant, but its use is not recommended where the mica is undercut as it will fill in the cuts and carbonize. Lint is harmful and care should be used that any cloths used around a commutator are fairly free from lint. Use no waste.

Brushes can be lubricated by boiling in oil, then let them stand to dry. A good way to apply oil for lubrication is to use the bare finger dipping first in oil

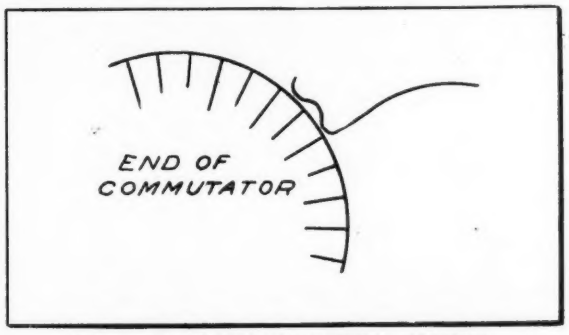


FIG. 6. EXPLORING FOR NEUTRAL POSITION

and then wiping the surplus oil off. Boiling brushes in paraffin is also a good way to lubricate them. When putting in new brushes see that the holder is clean and smooth inside, and that the brushes are just a nice free fit. See that the studs and brush holder are tight and in line and that the brushes set at the correct angle for the direction of rotation.

If brushes are set contrary to the direction of rotation there will be jumping, chattering and sparking. The number of bars divided by the number of poles gives the correct spacing of the brushes. To get accurate spacing put a strip of paper around the commutator underneath the brushes, keeping it tight, and mark the edges of all the brushes. Remove the paper and measure the distance between the edges of the brushes and also the width of the brush bearing surface. See that each brush has the same angle and contact face as the others. Sandpaper the brushes to fit the commutator by drawing a piece of sandpaper back and forth between the brush and commutator. While doing this have the springs under good tension and keep the paper conforming to the curvature of the commutator. Slightly round the edges of the brush faces but not enough to make any practical reduction of area. The brushes must be set in line lengthwise with the bars.

When a brush gets worn the tension spring hammer sometimes rests on the brush holder box instead of the brush and the brush will chatter. Brushes which pit and burn at the ends and glow along the edges are usually found to be of too high resistance or the machine is overloaded, or the brushes are of too small capacity.

Brushes should have as light a tension as they will run under to give satisfactory service. The tension will be a compromise between wear, chattering, sparking, and jumping and will depend on the nature of the work, the resistance and composition of the brush and

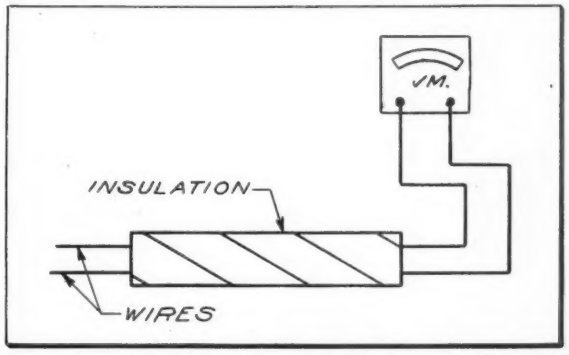


FIG. 7. USE OF VOLTMETER AND WIRE POINTS FOR FINDING NEUTRAL POSITION

whether there is vibration. The average tension in pounds per square inch in practice varies from 1 lb. and under for stationary work to 2, 2½, 3, or 4 lb., for high speed or large stationary work, and on up to 6 lb. for railway motors.

On account of its position and the force of gravity the lower brush often requires the most tension. See that the pigtails are not so stiff as to interfere with the free working of the tension spring.

Brushes should not cover too many bars or wobble in the holder. Very hard brushes are of high resistance and brittle, and will cut the commutator and themselves quickly and cause sparking.

Wax will often ooze out of the self lubricating brushes and cause them to stick in the holder or may coat the commutator. Pigtails are used to relieve the springs of carrying the full current. Brushes will break from a series of minute blows, inflicted by a slight unevenness of a high speed machine. Where more than one set of brushes carry current in multiple, see that the different sets carry an equal amount of the current. If not the good set will be overloaded and spark, pit and burn.

If a brush covers too many bars a strong local current may be formed, causing overheating of the brush.

When brushes appear to spark from overload try a softer brush of less resistance. Brushes which fit too tightly will not bear on the commutator. If they are too loose they will be thrown and chatter and spark. They will stick if not of correct size, if of uneven thickness, or from a coating of wax in the holder.

Remember that high resistance of a brush is desirable but too much resistance is bad. Boiling in paraffin will soften a brush and lower its resistance. Brushes are usually staggered so they will not all

bear on the same surface of the commutator, and cut grooves, and the shaft has end play for even wear. Some commutators are designed of such short length that any attempt to stagger the brushes and at the same time have the brush holder near the commutator will result in the holder grounding on the upright surface of the bars where the leads are soldered.

Sometimes a brush is set out over the edge of the commutator and a shoulder worn in the brush. In this case oscillation of the shaft will raise the brush and cause sparking. If one brush always sparks and the others are all right the spacing is probably not right.

Sometimes armature leads are carried a quarter turn before soldering to the bars and the brushes will then be set opposite the center of the pole face and not between the poles.

When a brush is too hard small particles or dust of copper will often imbed in its face. In this case lubrication is needed, or the copper is too soft. A brush may bear at only one point due to dirt, poor fit of bearing face, tight fit in the holder, etc. If the brushes chatter change the angle, or the tension or get a tighter fit. If the brushes squeak while under load change the grade or lubricate. Carbon brushes in good condition will squeak at slow speed when starting or stopping.

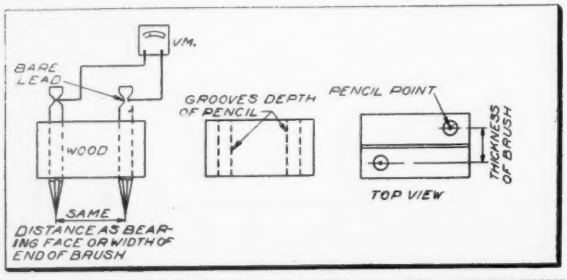


FIG. 8. USING PENCILS AND VOLTMETER FOR DETERMINING NEUTRAL POSITION

When dynamos spark from the mica or carbon being too hard there will likely be a coating of black smeared over part of the surface of each bar.

When setting brush holders there are 3 common methods in use for exploring for neutral. The simplest is as shown in Fig. 6. A wire is bent as shown of such a size that the 2 loops cover about the same distance as the width of a brush. By moving this to different positions and by raising and lowering the tip of the loop a place can be found where the tendency to spark is least. The next 2 methods require a voltmeter. One method is to take 2 wires as shown in Fig. 7. The other method is to take 2 pencils as shown in Fig. 8. The last method is called a dummy brush. The block with pencils in fits the holder the same as a brush.

The greatest portion of sparking may be traced to poor contact, giving the effect of high induction. The higher resistance may be due to small or loose conductor such as leads and pigtails.

General Nature of Sparks

SMALL, bluish, white sparks, usually extending all along the edge of the brush and continuous, are not particularly harmful and are caused by vibration, loose brushes, etc. There is a kind of fiery, generally red. sparks which fly some distance from the commutator which are not always harmful to the commutator, and which are generally caused by burning particles of carbon. The main loss in this case is wear of the brush.

Green, yellow, or bright red sparks are vicious, snappy, and "hot," their color being due to burning copper. They cause intense heat and damage the copper and brushes, often suddenly putting a machine out of commission. If let run too long this way, the mica will burn and char and the solder will be melted from the armature leads. An open-circuited armature will cause violent sparking at regular intervals, burning bars, and pitting and burning the mica. The remedy is to solder the leads together or use a jumper. In case of brushes sparking from vibration try a little more tension on the springs.

Oil rots mica; oil also becomes carbonized; be careful of oil running over from the bearings.

A varying or sudden heavy load causes the current to "bite" by induction, into the copper. Oftentimes one brush out of a set is not working, causing others to do too much.

Roughness of the commutator may be detected by feeling lightly with a pencil point or the finger nail.

Generally a spark staying in one place is due to dirt or copper chips imbedded in the brush.

Always be careful not to have dirt or dust on machines. When not running, give them a good blowing with bellows or compressed air at not over 20 lb. pressure per square inch. This treatment is especially needed after truing or using sandpaper. Faults due to poor brush holder design, field design, springing shaft, etc., are up to the maker.

General Care

WHEN machines do not build up, but spark at the brushes, if the field is all right the trouble is likely to be poor commutation. If you haven't time to fix the commutator before the load comes on, tie a piece of fuse wire, not over ¼ the capacity of the machine, to the positive and negative terminals. As a sudden building up of current melts this wire and causes it to fly, a rag should be wrapped around it to prevent the melted metal from flying on the face and eyes. When attaching the wire be sure the resistance is all cut in the field and that there is no voltage on the machine. After attaching, slowly open the rheostat. This method is only recommended for machines of small capacity and in cases of emergency. No machine should be allowed to get in this condition when possible to avoid it. It is sometimes unavoidable with machines in constant use.

Picking up can sometimes be hastened by bearing on the brush. It can sometimes be hastened by bearing lightly on the commutator with fine sandpaper, as there may be a thin film of wax on the commutator which has an insulating effect when cold. The method of using a piece of fuse wire as mentioned above is very good in some cases where the residual magnetism is weak. Keep moisture and injurious fumes from the machine. Wipe the commutator daily with a dry rag, taking care that no lint gets on the commutator. Never use waste.

Examine all leads frequently, as loose connections cause higher resistance and by jar cause induction. If the commutator gets gummy and dirty take a slight amount of kerosene on a rag or the finger to remove the gum, first removing the brushes so they will not soften and smear the commutator. After dipping a rag in kerosene draw it across another dry rag to remove the excess oil.

Weak field density allows more field distortion from armature reaction. One remedy sometimes used by shop men is to remove some liners from the back of the poles. This makes a heavier density needed to give a proper strength to the field for the same armature load as before. As this density is obtained by a heavier field

current it is more resisting and does not shift so far for a given armature reaction. The armature reactive force remains constant for a given load.

Too much load causes sparking because of the resistance of the brushes and also on account of too much armature reaction. There are mechanical causes of sparking, such as warm bearings, hot frames, dirt preventing radiation, etc. Rise of temperature in a room affects the machine the same as overload by increasing the resistance of the conductors, field coils, brushes, etc. There are causes of sparking outside the scope of this article such as design, but these concern the designer and builder rather than the operator.

WESTINGHOUSE WATERWHEEL GENERATORS

WATERWHEEL generators have recently been built to conform with the rapid development of hydraulic power for driving electric generators. Two types, horizontal and vertical, depending upon the local conditions in each case, of almost any practical capacity or speed for installation in the smallest isolated plant or the largest hydroelectric generating station have been furnished by the Westinghouse Electric & Mfg. Co.

FIG. 2. LAMINATED RIM ROTOR, SHOWING SPIDERS, AND GROOVES FOR FASTENING FIELD POLES

The standard horizontal unit is of the 2-bearing, coupled type construction; that is, the generator includes shaft, 2-bearings, and a bedplate usually designed to allow for sliding the stator to one side in case ready access to either the stationary or rotating winding is desired.

The stationary frame is made of a strong, rigid iron casting, into which soft steel laminations are dovetailed and securely fastened.

Ventilating ducts are spaced at frequent intervals across the face of the armature punchings, allowing for perfect ventilation to all parts of the active material.

Form wound, interchangeable armature coils fit into parallel open slots punched in these laminations, and these coils are held firmly in place by means of fiber wedges. The coils are insulated and impregnated with fabrics and compounds of high insulating qualities.

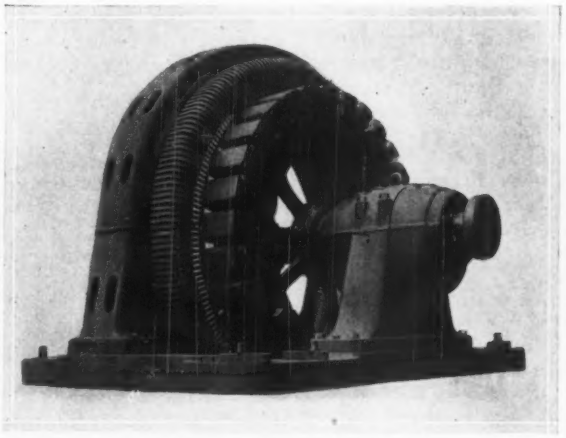


FIG. 1. WATERWHEEL GENERATOR COMPLETE, SHOWING METHOD OF SLIDING STATOR TO ONE SIDE TO UNCOVER ROTOR

No single type of construction will meet the varied requirements in rotor design, therefore, several well tested methods are employed. When comparatively low peripheral speeds are encountered a cast-iron spider with bolted-on, or dove-tailed poles, is usually employed. For higher speeds cast-steel, or steel plate

FIG. 3. COMPLETE GENERATOR OF VERTICAL TYPE, 12,500 KVA., 6600 VOLTS, 50 CYCLES

construction may be used. In the case of very large relatively high-speed machines, the difficulty of securing perfect castings may lead to the well-proven laminated rim structure.

All field poles are made of thin steel laminations riveted together with overhanging pole tips provided to support the field windings. Field coils are wound of heavy copper strap on edge, insulated in such a way that each individual turn is exposed to the ventilating air, and thus perfect radiation results. The coil is securely fastened between the rotating spider and the tips of the field poles by heavy coil supports.

When completed, the machine is carefully tested under conditions, as nearly identical as possible, to those which its future service will demand.

Westinghouse standard practice recommends that the vertical type generator be fitted with 2 guide bearings which are supported by brackets fastened to the stator frame. Also, a bedplate or pad on which the stationary part rests.

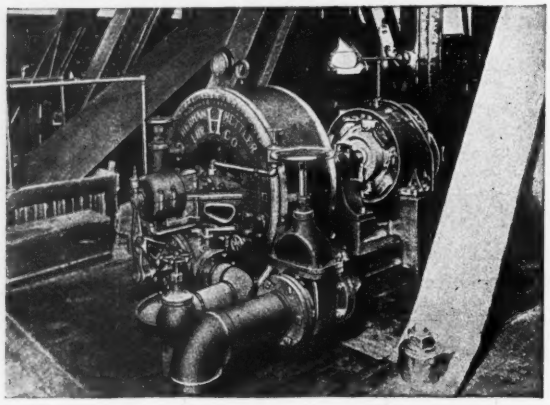
The roller, or thrust bearing, which supports the weight of the revolving part may be mounted on top of the generator frame between generator and turbine, or underneath the turbine. In case it is mounted on top of the generator frame, this frame must of course be made heavier and more expansive, than where it has only to support the stator punchings, winding and guide bearings. Wherever placed, this bearing usually supports not only the rotor of the generator, but also the turbine runner, and in addition takes care of any unbalanced water thrust.

In case of failure of a waterwheel governor to act, the rotating part of both waterwheel and generator are subjected to unusual stresses, due to the overspeeding of these parts. The rotors here illustrated are designed for the maximum overspeed varying from 50 to 100 per cent.

THE SMALL TURBO-GENERATOR

Increasing Use as a Relay Set

USE of the small turbine-driven generator for relay purposes is a phase of modern power plant practice that is continually increasing. The ability of the turbo-generator set to make long continuous runs with little attention, its reliability, and the fact that it can be started from a cold condition in a few minutes, are points which have warranted its adoption.



TURBO-GENERATOR RELAY SET

As illustrative of how a turbo-generator set is being used for relay work, the case of the Hettler Lumber Co., Chicago, may well be cited. This concern makes a specialty of hardwood flooring and furnishes a large part of all the flooring used in and shipped out of Chicago. The prime mover at this plant, a large Corliss engine of about 350 i.hp., drives the mill by belt to overhead shafting. The mill shuts down between 12 and one o'clock each day and during that time the 50-kw. Terry turbine-driven generator set is used for lighting and power purposes. At 6 p.m. the turbine is again started and operated until 7 or 8 o'clock the next morning. During the night it supplies current for watchmen's lights, dry kiln fan motors and for emergency use in case of fire in the lumber yards. The mill is obliged to burn a certain number of arc lights in its yards all night in accordance with the rules of the Chicago fire department. The interior must also be electrically lighted at night as watchmen's lanterns are strictly prohibited.

As the turbine exhaust is free from oil, it may be utilized directly in the dry kilns. All detached machinery is driven by motors, the dry kilns requiring a constant supply of clean steam piped through coils, and the drying is accomplished in the usual way by means of variable-speed motor-driven fans. The main shaft, which is supported on A-frames, runs the length of the planing mill, a distance of about 350 ft., with planers, matchers, rip saws, etc., are belted down from this shaft.

WHY THE MOTOR HAD 2 WIRES

By J. J. Pire

ONE day while peacefully contemplating the achievements of the past and speculating on the probabilities of the future, comes in a man with a real problem. The given quantities were these: One hall of large dimensions with a floor which was to be sandpapered. Unwillingness to do the job by hand, and a 1 hp. gas engine available.

The desired answer was a dynamo connected to the engine, flexible leads and a motor-driven sandpapering machine. The missing quantities were dynamo and sandpapering machine, but the latter with its motor were possessed by another hall owner and my "client" guessed it could be borrowed, leaving only the dynamo to be provided.

There was, however, the "if" in regard to the motor, so my man said: "Well, I'll go see about it," and I told him to note the power of the motor to see whether we could run it.

Meeting him later, still hunting for a dynamo, I asked: "But what was the power of the motor?"

"Well," he answered, "that's a pretty big motor. They have to use 2 wires to run it as it needs 200 volts and with one wire they can't get over 100 volts and she won't work on that." That was all I could get out of him, and seemed to be all he knew, but he was very certain of that much, so I decided to look it over myself.

I went, expecting to find some freak design double cross-compounded, noncondensing, triple expansion unipole or differential motor, but to my surprise found an ordinary direct-connected shut wound motor for 200 volts, which they had been operating with double leads instead of a single duplex flexible.

But on the name plate was plainly stamped 3.5 hp. This, in view of the 1 hp. gas engine made 2 missing quantities, a dynamo—and power, so the problem was marked as unsolvable even though the "one wire motor" was no mystery.

CORRESPONDENCE OF AN OLD ENGINEER AND HIS SON

Bluffers in the Power Plant

MY DEAR SON DONNY:

Your letter of the 18th was received and we were all very glad to hear from you again. It is a long time between letters but I suppose that it is an evidence that the time is not as plentiful as it ought to be. Yes, I know all about it for I went through the mill myself 50 yr. ago,—and more.

I know what it means to stay after quitting time and pack the piston with hemp the way we had to do in those days, and tinker with those old strap end rods that seemed to defy close adjustment. Still, it had to be done, and there was no one else to do it but the engineer.

Then, in those days I had the girl fever just the same as you have now,—and I have it still. Your mother was one of the belles of the city then and I still think so. Judging from the little remarks that slip into your letters from time to time, I guess that there is a "belle" in Daneberg also.

I am glad to see that you are using your spare time in keeping ahead of the game, by studying up a little. This life is a lively proposition at best, and it is about all that we can do as a general thing to keep with the procession, to say nothing of being in it or at the head of it.

Did you ever see a man at the head of anything —and staying there—who put in his evenings ramming around the town looking for excitement? Today is the time of all times when a man has got to keep right on the job if he is going to come out ahead. Some fellows get along pretty fair by "running a bluff," but they are never what you might call a real success.

When a fellow comes on a job and talks a lot the first thing, he hasn't got much to say all together. That is like a fellow that came into an engine room several years ago where they wanted a night engineer. The plant was a small one consisting of a cross compound Corliss direct connected to a generator of about 250 kw. capacity. This young fellow began to tell the engineer in charge, who it seems had the authority to hire, all about what he knew, then about the good jobs he had. He was 2 yr. here and 3 yr. there; then he ran 4 yr. at another place and resigned (he didn't say, "got fired") to accept a position as night engineer 1 yr. at some other place so he could have his days to study, and quit there and went to another firm for 2 yr., days, so he could go to night school, and so on.

This got on my nerve, so taking out an old envelope and a pencil I jotted down the years he worked and they totaled about 18 or 20. Well he was hired, and went off to get a place to sleep, and I asked the engineer if he had asked the young fellow's age, and he had. It was 27 yr. I added up the figures again to be sure, and then we calculated that this mechanical phenomenon either had begun to be an engineer at about the age of 8, held several jobs at the same time, or else was a disciple of Ananias.

Well, he came to work that night all right and after the day engineer had showed him all about the switches and valves amid several oh,—that-don't-bother-me-a-bit expressions left him with his charge.

Next morning there was a carefully prepared list on the table, of the different things that ought to be fixed according to his view, and they ranged all the way from putting grease cups on the slides to moving the switchboard.

He was somewhat like another fellow who had lots to say most of the time. This man was a good talker but there wasn't a governor on his vocabulary. He used to say things on the spur of the moment, just to hear the noise,—and they were amusing sometimes.

He was one of the engineers at a place where they had several engine rooms which had been built before the days of electrical distribution, and they built the power houses where they would be central to some one or more departments.

In one of these engine rooms they had a vertical Corliss engine which did the unusual stunt one day of losing a crosshead shoe. I do not recall now whether the nut came off or broke off, but anyway, the shoe came off and flew up and caught over the flange of the stuffing-box gland where it hung out of the way, so much so that it wasn't found for some little time.

This fellow, Harris, came in just ahead of the chief who inquired as to the shutdown. The engineer of that particular engineroom, said to the chief, "The engine has thrown one of her shoes."

Harris wanted to be agreeable without knowing how, so he volunteered the startling and unusual information,

"Oh, they often do that."

"Well, if they do, I'm going to quit," said the engineer.

So, after looking at this matter for about 75 yr., I have come to the conclusion that a young man had better cull his girl acquaintances down to one, and then win her by showing her that he is the proper one to earn a living for her. That is the way I did with your mother 51 yr. ago, and if I had it to do over again, I wouldn't change the process.

Well, mother says that I must empty a bag of flour so she can set bread, so I will close for this time. Write when you can get time, but don't let the oil cups run dry getting it.

Affectionately, your father,

Sandy MacDougal.

IN DECIDING THAT the city of Goshen, Ind., has a right to repair or rebuild its municipal light plant without seeking authority from the state, the Public Service Commission ruled that it has no injunctive power and that the utility act does not give the commission power to prevent an owner of a utility from extending or repairing its plant, even though such extensions or repairs amount in effect to the building of a new plant, if the protest against such extension is made upon the ground that wasteful competition will ensue. The Hawks Electric Co., of Goshen, a private corporation, had sought to prevent the municipality from rebuilding its wornout plant.

"If the city is preparing to string new wires only or set poles or install a new dynamo its right to do so would never be questioned," said Commissioner Clark, who wrote the commission's order. "Because it seeks to do all these things at once, no doubt in the interest of economy and better service, this fact does not make the statute applicable. It could not have been the intention of the Legislature to compel municipalities to continue to use antiquated machinery or abandon it altogether, if the certificate of convenience and necessity could not be procured from this commission to install new and modern appliances.

STEAM METERS*

Installation, Care and Maintenance

THE following rules for the installation and maintenance of Simplex meters have been furnished us by the manufacturer.

It is of the highest importance that the meters should be of the proper capacity, the correct sizes being determined by definite rules, otherwise at maximum periods of flow, an accurate registration would not be obtained.

The proper size of meter required for all ordinary installations may be ascertained by dividing the number of square feet of radiating surface in the building by 0.33. The figure thus obtained will correspond to the average maximum flow in pounds of water per hour.

The proper size of meter may then be selected by reference to the following table:

NUMBER	2	3	4	6	7	10
Capacity of meter per hour, lb. water	150	375	750	1500	3000	15,000
Inlet connections, inches	¾	1	1¼	1½	2–1½	2
Outlet connections, inches	2	2½	2½	3	2-4	4-4
Capacity each double dump, lb. water	2	5	10	20	40	200

Directions for Setting

THE inlet opening is on top of the meter and 2 outlets are provided, one in each end of the meter case, either or both of which may be used.

The meter must be set level on a rigid foundation. It should also be installed with unions to permit of easy removal for inspection and testing. All meters, wherever possible, should be located where they may be easily read.

A series of cast-iron economy coils should be placed between the steam trap and the meter. The coils should have a radiating surface of not less than 20 per cent of the total radiation in the building. The coils should be enclosed in a tin lined box having a register in the floor above. The outlet from the coils should be carried up just above the highest point of the coil in order to form a water seal, and from this level the condensation should be provided in the pipe between the coil and the meter to prevent syphoning the coil.

The outlet of the meter must be located above the sewer to insure a free flow by gravity from the meter. The meter should be installed as near to the sewer outlet as possible and the discharge from the meter should be piped directly to the sewer. The pipe should be as short as possible and grade rapidly away from the meter. Never bush or reduce the size of the meter outlet and never reduce the size of the first 10 ft. of discharge pipe, leading from the meter. If the total length of the discharge pipe is less than 10 ft., do not reduce the size of the pipe.

A sewer trap should in all cases be installed to prevent water or sewer gas backing up into the meter.

Where the "Atmospheric" system is installed economy coils are not required, a "Receiver" performing the same function in connection with proper meter operation. In order to obtain the best results economy coils and a receiver should in all cases be installed ahead of the meter.

We strongly recommend the adoption of the "Atmospheric" system of steam heating for all new installations and a proper amount of economy coils for all other systems of heating so that the condensation may be cooled to approximately 85 degrees F., the consumer thereby saving from 10 to 15 per cent in the cost of heating, assuring to the consumer as well as the company, the most perfect meter operating conditions. The central station heating companies should make this an invariable condition.

The "Simplex" condensation meter should not be used (where pressure systems are installed) in connection with traps which have intermittent discharge or permit the escape of steam through the outlet.

Condensation Meters in Service

A CAREFUL inspection of the meter is one of the most important duties of the meter reader, who should note that the

Meter sets level.

Meter seals are intact and no evidence of tampering.

Movement of bucket is not impeded in any manner.

A most important feature of a central heating plant is the correct installation and operation of its meters. Meter readings should be taken approximately every 10 days. In very large buildings, we recommend that more frequent readings be taken, particularly in new installations. It will be understood that the frequent readings are taken for inspection purposes, the bills being rendered on the total monthly consumption.

The meters should be cleaned at once, whenever it is found to be necessary.

Should a meter become clogged from sediment, etc., remove the cover and loosen the screws which fasten the bucket to the shaft. The bucket may then be removed. In replacing the bucket, observe the word "Front" stamped on the bucket, to be sure that the bucket is replaced in its original position.

A small lever on the back of the meter cover is provided for cleaning the inlet nozzle. This lever should be worked back and forth a few times when the meter is inspected.

The inlet of the meter is relatively small in order to prevent flooding.

Typical Routine of Operation in a Meter Department

MR. HIGGINS, of Detroit, has given us the following routine which is used in reading meters for the Murphy Power Company. Mr. Higgins' method is as follows:

We will assume that Frank Jones has made application to the company for service. After the contract has been signed and approved, a record card is made out, giving the name and address, type of building, square feet of wall surface, square feet of glass surface, and cubical contents, also the rate at which the contract is taken. There is also a card sent to the meter department which calls for installation of service, with size of meter, and there is a space below on the card to be filled out, stating when the meter is connected and reading of the meter. This card is made out in triplicate. The original and duplicate are sent to the

*Continued from page 895 Sept. 15 issue.

meter department, and the triplicate is kept in the office for reference in case the installation of the meter is delayed. When the meter is installed, the original card is returned with the information as given above. This card is turned over to the accounting end of the meter department.

Another card is made out which shows the name, address, number of meter and size, and has space provided for the season's readings, one column being for date, and the next column for readings of the meter. Now the meter is in operation.

Meters are read at least twice a week, although in some cases where the customer objects they are read less often. The meter readers do not really read the meter, but simply have a card on which is marked the name of the customer and the dials of the meter. The meter reader's duty is simply to see that the meter is working properly and to copy the dial face correctly on the dial slips. An exact reproduction of the dial needles and where they point is all he is supposed to know. This is done in order to prevent the meter readers having possession of previous readings and making up readings without looking at the meter. When the day's readings are done these dial slips are turned in to the office for record.

First, the dial slips are sorted and arranged alphabetically, then they are read and the reading, or interpretation, put on the bottom of the slip. After the readings have been interpreted the dial slips are passed on to another, who copies the readings on the permanent card above referred to, and which carries information as to the performance of this meter while installed on the premises of this particular customer. If the inspector, or reader, notices anything the matter with the meter, such as returns leaking or meter stuck, it is noted on this card. In fact, anything out of the ordinary is put down here so that the card gives a complete history of the performance of this meter during the entire period. Sometimes you will find on the card a period of a few days where the meter shows no consumption, and in the margin is marked T.C. This means Trouble Card. In comparing current reading with the previous one it was found that the meter was not registering and a trouble card was sent to the meter department asking the reason.

This card is made out in triplicate. The original and duplicate are sent to the meter department, while the triplicate is kept in the office for reference. When the trouble is located and the meter repaired the original card is returned to the office. If the meter is found so defective that it is necessary to bring it in for repairs and a new meter substituted, a change record card is made out and sent to the office. This card shows the last reading of the old meter and the first reading of the new meter. This makes a complete record of transfer of meter on customer's card and the system of reading goes on as before.

The meters are tested once a year unless the customer requests otherwise. In testing the meter a complete record of the test is sent to the office for filing. This meter test card is marked with the number of the meter, the customer's name and location, and the condition before cleaning and after cleaning. There is also a meter record card which is filed numerically and shows the change of individual meters from place to place, and is a history of the meter only.

At billing time all permanent cards with billing readings inscribed are put on the adjuster's desk. He follows out the consumption in each individual case, and where the meter has been stuck or has failed to register for any reason whatsoever, an addition is made to the consumption, plainly marked on the bill when going out, and a letter written to the customer explaining in detail what was done and why it became necessary to do it. These estimates are based on the average consumption during the period when the meter was working properly, with an allowance for outside temperature. After these cards have passed through the adjuster's hands and been sent to the billing department they are complete and ready for making out bills. In other words, the billing department is relieved of all responsibility. They simply copy the monthly consumption as shown on the card into the ledger and make out the bill.

This record card which is sent to the meter department is filled out in 3 colors of ink. Black is used for ordinary readings; red for the monthly, or billing, readings; and green for the monthly consumption, or the difference between the 2 consecutive monthly readings.

Before these cards are sent to the billing department the readings are O. K.'d by the adjuster with a little stamp.

The above is the routine followed by a company which has a large number of customers and it is evident that the greatest care is taken both to prevent error in readings, and to prevent false reports being made by meter readers. It is possible that some of these precautions might be found unnecessary by a larger company. For example, it might be well to follow the practice of most companies in reading electric meters, to give the meter reader the opportunity of knowing what the previous meter reading is in order that he can check the reading which he takes with the previous reading, the assumption being that if the meter reader is honest, it will be an additional help to him in giving correct readings if he is able to check with previous readings.

THE LARGEST POWER PLANT in York County, Maine, is to be built by the Pepperell Manufacturing Co., at Union Falls on the Saco River. The plans call for a concrete dam 70 ft. high and the development of 10,000 hp. The building of this huge dam will cause the water to flow back into the country for miles and will flood sections of 2 highways, some parts of which will be under 40 or 50 ft. of water when the great reservoir which will be created has been filled. It is proposed to locate the dam near or on the site of the old covered bridge just below the present dam at Union. The estimated cost for a plant to develop 10,000 hp. is placed at $1,000,000.

Representatives of the Pepperell company have been engaged in purchasing flowage rights in the region for a long time, and at present about all such land that will be required has been bought.

On Cook's Brook is a water privilege owned by Senator Cecil F. Clark of Clark's Mills. The flowage of the region will destroy this power, so it was purchased by the Pepperell company, and Mr. Clark was allowed to locate a turbine at Union Falls, where the power plant of the Clark Power Co. is located. This company has now been bought by the Pepperell company. It was capitalized at $10,000 and supplies electric lights to the towns of Hollis, Bar Mills, Salmon Falls and Clark's Mills. Its turbine at Union develops 400 hp.

The work will require a long time for its completion. How soon the work will be entered upon cannot be stated, as parties conversant with the matter state that it may be 2 or 3 yr., or much sooner.

POWER PLANT CHEMISTRY

Using the Flue Gas Analysis Instrument to Aid in Perfecting Combustion

IN building up the efficiency of a plant the first step is to make sure that the boiler furnaces are working at the highest possible efficiency. To determine this there are several methods, but the most simple and one which can be used by any engineer, is by means of the flue gas analysis. Ultimate analysis of the gas is not necessary, the determinations commonly made are the percentages of carbon dioxide (CO_2), oxygen (O) and carbon monoxide (CO).

Knowing these, it is quite easy to ascertain whether the combustion of the fuel has been complete, for if the gas contains CO_2, a percentage of O and no CO, it is evident that all carbon has been completely burned, but it is quite probable that an excess of air is being furnished the fuel so that its temperature is reduced more than need be. By reducing the air supply, the percentage of O will be reduced until, theoretically, an analysis of the gas should show 20.91 per cent CO_2, no O and no CO, which would be perfect

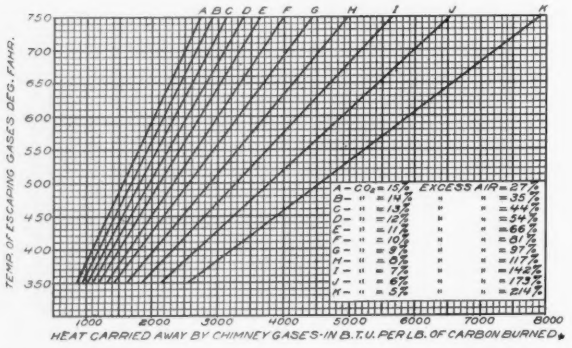

TEMP. OF ESCAPING GASES DEG. FAHR.
HEAT CARRIED AWAY BY CHIMNEY GASES IN B.T.U. PER LB. OF CARBON BURNED
A B C D E F G H I J K
A-CO2=15% EXCESS AIR=27%
B- " =14% " =35%
C- " =13% " =44%
D- " =12% " =54%
E- " =11% " =66%
F- " =10% " =81%
G- " =9% " =97%
H- " =8% " =117%
I- " =7% " =142%
J- " =6% " =173%
K- " =5% " =214%

FIG. 1. LOSS DUE TO HEAT CARRIED AWAY BY CHIMNEY GASES. BASED ON BOILER ROOM TEMPERATURE OF 80 DEG., NITROGEN IN FLUE GAS 80.5 PER CENT, CARBON MONOXIDE 0 PER CENT. (COURTESY B. & W. CO.)

combustion. This condition, however, is never attained in the power plant, best results being obtained, all things considered, when the air supply is 50 to 100 per cent more than theoretical requirements. A reduction of the air from the above figure is likely to cause incomplete combustion and the formation of more or less CO in the flue gas accompanied also by smoke.

The problem is, then to maintain such a condition in the furnace that the percentage of CO_2 will be high and the O as low as it is possible to maintain it without giving more than a trace of CO.

The gas analysis which gives best results varies with the design of the furnace and the quality of coal. As a guide, however, the following table may be used in a general way where complete combustion is obtained, which means that no CO is formed.

CO_2 in per cent	O in per cent	Air excess in per cent	Fuel loss in per cent	Preventable Fuel loss in per cent
12	8.9	72.5	15	3
10	10.9	107.	18	6
8	12.9	158.7	23	11
6	14.9	245.	30	18
4	16.9	417.	45	33

*For loss per pound of coal multiply by per cent of carbon in coal by ultimate analysis.

Having an ultimate analysis of the coal, the approximate quantity of air required for complete combustion may be calculated as follows:

$$A = 34.56\left(\frac{C}{3} + H - \frac{O}{8}\right).$$

in which

A = weight of air required per pound of fuel;

C, H and O = the proportional part by weight of carbon, hydrogen and oxygen in the fuel.

Also when the analysis of the flue gas is known it is possible to calculate approximately the weight of air supplied to the fuel by the following formula:

$$A = 3.036\left(\frac{N}{CO_2 + CO}\right) \times C$$

in which

A = weight of air supplied per pound of fuel;

N, CO_2 and CO = percentages by volume of nitrogen, carbon dioxide and carbon monoxide.

C = the proportional part by weight of carbon in the fuel.

It is usually quite safe to assume that the percentage of nitrogen in the gas as equal to the difference between 100 and the sum of the percentages of CO_2, O and CO.

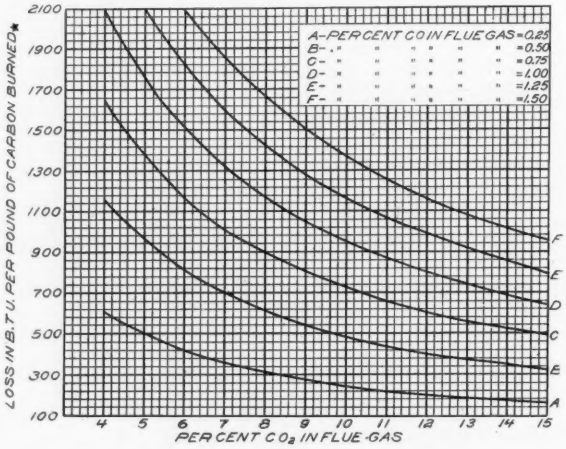


FIG. 2. LOSS DUE TO UNCONSUMED CARBON CONTAINED IN THE CO IN THE FLUE GASES. (COURTESY B. & W. CO.)

From these 2 formulas, then, we are able to determine the deficiency or excess in air that is being supplied to the fuel and thus we get a measure of the furnace efficiency.

The heat lost in the flue gases is $L = 0.24\ W\ (T-t)$, in which L = B.t.u. lost per pound of fuel;

W = weight of flue gases in pounds per pound of dry coal;

T = temperature of flue gases;

t = temperature of atmosphere;

0.24 = specific heat of flue gases.

To compute the weight of the flue gases per pound of dry coal from the gas analysis use the following formula:

$$W = \frac{11\ CO_2 + 8\ O + 7\ (CO + N)}{3\ (CO_2 + CO)} \times C$$

in which CO_2, O, CO and N are the percentages by volume of the carbon dioxide, oxygen, carbon monoxide and nitrogen in the flue gas, and C is the percentage of carbon in the coal as determined by ultimate analysis.

Having once determined the conditions of the furnace which gives the best gas analysis it is reasonable to assume that, for the load being carried, the draft gage is then indicating the most economical draft, and the fireman should be instructed to carry it at that point.

When the flue gas analysis shows a comparatively large percentage of oxygen and at the same time carbon monoxide, it is conclusive evidence that there are leaks in the boiler setting which do not compel all the air passing to the stack to go through the bed of coal. To locate these leaks, the gas analysis instrument can be used to advantage by taking samples at various points along the path of the gas travel, a marked difference in the analysis of gases from points a short distance apart will indicate closely the point of air leakage.

In many cities smoke ordinances are enforced necessitating more care in the operation of the furnace than where efficiency alone is to be considered. In these plants it is customary, in order to be on the safe side, to furnish the fuel with a greater excess of air and make sure that combustion is complete. While smoke is caused by free carbon in the flue gases, it is not a gas and the ordinary analysis of the gas will not determine its presence. A fair idea of the amount of smoke may be had from the percentage of CO in the gas. The following from a bulletin of the U. S. Geological Survey shows this clearly:

Average per cent of black smoke:

0	7.1	15.5	24.7	34.7	43.1	52.9

Average per cent CO in flue gases:

0.05	0.11	0.11	0.14	0.21	0.33	0.35

As stated in a previous article the flue gas analysis is an accurate indication of the quality of combustion which has taken place in the furnace. The analysis in itself cannot improve furnace conditions but by a series of experiments with different methods of handling the furnace and a careful study of the flue gas analyses an engineer can determine what conditions in the furnace give best results and may discover defects in the furnace or setting which can be remedied without great expense. It is possible also to find out whether the coal being used is that best adapted to the furnace. As an analyzer of furnace difficulties the flue gas analysis instrument is an indispensable accessory.

NOTES ON LUBRICATION

As a result of tests made on cup greases for lubrication, A. L. Westcott states in a paper before the American Society of Mechanical Engineers that rise in temperature results in lessened friction and that the effect of rise of temperature on the coefficient of friction is nearly always the same. The reason is that fluidity of the lubricant is increased by warming it and viscosity is decreased, which results in decreased friction up to the point where the bearing pressure overcomes the tenacity of the film of lubricant, and forces the rubbing surfaces into contact. There seems to be no danger of cutting the bearing up to 150 deg. F. as long as the film of lubricant holds, and even 200 deg. was reached satisfactorily as long as there was ample supply of lubricant. This goes to show that there is nothing objectionable in a temperature of bearings much higher than commonly permitted, if good lubrication is maintained, and much may be gained in the way of decreasing work lost in friction.

Curves in the Data Sheets show the relation found between temperature and coefficient of friction for one set of tests on a cup grease; also that increase of unit load on the bearing reduced the relative loss from friction with the bearing held at a given temperature, because the friction is that of particles of lubricant on each other, so long as the film is not broken, and increase in load carried up to the breaking point of the film does not increase, proportionally the force required to slide the lubricant particle.

As a result of his tests Mr. Westcott concludes that:

Grease lubrication compares favorably with oil where the form of bearing is such as to favor the retention of the film of lubricant, and provision is made for an ample supply to the bearing. But, as shown by the experiments, oil may give better results in case of a short bearing in proportion to the diameter.

Grease of soft consistency is a much better lubricant than the harder densities of the same grease, an advantage especially marked at low temperatures, such as usually obtain in a well lubricated bearing.

The best method of applying grease to a bearing is by forced feed and a constant rate of flow. This agrees with the best practice in oil lubrication where the bearing is flooded with oil, which passes to a filter and is used again. The problem in case of grease is to clean it after it has once passed through the bearing so that it can be used again.

Some grease cups with spring actuated plungers are designed to give a constant flow of grease, but when full of grease, so that the spring is compressed to its fullest amount, the pressure upon the grease is greater than when the cup is nearly empty. The experiments upon grease consistency show what a great difference in flow is produced by a small change in the pressure upon the grease. Provision made to regulate the flow by means of a small cock placed in the outlet of the cup, needs adjustment as the cup empties, and is apt to be neglected. A design of cup is desirable which will deliver the grease at a constant rate from the time it is filled until it is empty.

At a recent meeting of the Minneapolis Association of Stationary Engineers, State President Mueller, chief engineer of the Washburn-Crosby Co., who visited Germany and other European countries with the American Society of Mechanical Engineers this summer, gave an interesting and instructive talk in regard to his trip across the water. Following this talk he was presented with a gold watch charm by the association in appreciation of his efforts during the past year as chairman of the Educational Committee. The association was presented, by Peter Ulstrom, chief engineer of the C. A. Smith Lumber Co., with a hand carved gavel which he had made during a recent sojourn in Oregon. Some discussion was held of the second annual banquet, but definite plans were laid over until the next meeting. Following the meeting, some 75 of those present accepted the invitation of State President Mueller to an informal lunch which was heartily enjoyed.

The Eastern Pennsylvania Power Co., Easton, Pa., is preparing plans for alterations and additions to several large power plants.

FOREIGN DEVELOPMENT IN THE POWER PLANT FIELD

Roller Wheel Worm Gear, Heat Value of Fuels, Metal Working

By J. H. Blakey

ACCORDING to Unwin the efficiency of a single thread worm gear varies from 25 to 40 per cent; double thread, from 40 to 60 per cent; triple thread, from 50 to 66 per cent, and a quadruple thread gear under the best of conditions never reaches higher than 80 per cent. Consequently, a single thread roller wheel worm gearing which gives by test an efficiency of over 90 per cent will be of interest to engineers. The accompanying cut shows this speed reducer, which is manufactured by Glaenzer & Co., of Paris. The report of the government inspector shows that 3 tests were made by means of an electric motor and a prony brake. The voltage was 115, the amperage 114, the efficiency of the motor being 0.87, running at 1036 r.p.m. The power developed in the motor shaft was therefore:

$$\frac{115 \times 114 \times 0.87}{736} = 15.49 \text{ hp.}$$

The power transmitted by the gearing was found to be 14.46 hp., which is equal to 93.34 per cent. The temperature of the bearing on the motor side showed an increase of 31.5 deg. C. and on the other side an increase of 23 deg. after running 5 hr.

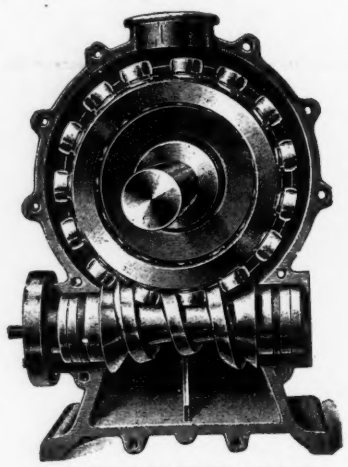
Influence of the Temperature of the Flame on the Value of Combustibles

COST of combustibles, whether in the form of coal or otherwise, forms such an important element in manufacturing that exact ideas with regard to fuel values are necessary. Whenever a combustible is to be chosen the following question presents itself: What is the exact value of this or that combustible in this particular case? It is generally the custom to estimate the value of a combustible simply by its calorific power, but this method is not exact and it is necessary to make use of closer calculations. The influence of the temperature of the flame must be taken into account, without losing sight of the fact that a certain volume of combustible of given calorific power cannot be economically replaced by a larger volume of combustible having a greater calorific power.

In "Stahl und Eisen," M. Gwigner considers various cases of the practical use of the principal industrial combustibles, such as coal dust, coal in the usual form, blast furnace and coke furnace gas, and gases of other kinds. He arrives at the following conclusions: Taking as a basis of comparison the ordinary combustion of coal in a grate, the author finds that for combustibles of equal calorific power per unit of weight or volume, that one is the most economical which gives the highest flame temperature. In this respect coal dust is superior to all others. After that, coke furnace gas is most economical. Air gas (ordinary air containing vapor of petroleum, naphthaline, gasoline or other hydrocarbon) in relation to the unit of weight of coal producing the same volume of gas has a heat value greater than that of the coal burnt normally in a grate, when the price of coal and labor is high. Next in value come the compound gases, such as blast furnace and coke furnace gases. Coal of such size as to require a large excess of air for combustion, even though of high calorific value, may have a useful value smaller than that of coal of the usual size, but of lower calorific power. Blast furnace gas is considered the least economical of all the combustibles, and its enrichment by means of regeneration in a gas producer does not appear to be profitable.

Effects of Cold Working Upon Iron and Steel

DIFFERENT methods of the cold working of iron and steel in the workshops in order to give them definite form, produce modifications in their structure and also in their physical properties which may have considerable importance from the point of view of their final use. In "Stahl und Eisen," Mr. Goerens describes the experiments which he has made on Martin steel and iron wire of different hardness, which he submitted to cold drawing in order to study these modifications. The author gives curves showing the elastic elongation and permanent set of the wires, which differ considerably according to the amount of drawing to which the specimen has been submitted. He then describes the effect of the drawing upon the limit of elasticity, upon resistance to rupture by extension, upon hardness, density and magnetic properties, and finally upon the resistance to solution in acids.



A NEW FORM OF WORM GEARING

The greatest modification was found to occur in the magnetic properties of the wire, and it was found that all changes were much more marked in the first drawings than in those later. The author expresses the degree of deformation by the relation of the diminution of section of the drawn wire to that of the original, and establishes the fact that the variations in the physical properties of the wire are always greatest for degrees of deformation between 0 and 50 per cent, and that they remain almost constant when the deformations exceed 90 per cent.

The Magnolia Power Co. proposes to construct a hydro-electric plant developing 13,000 hp. on the South Branch of the Potomac at Springfield, W. Va. The total cost will be $1,000,000. It is proposed to transmit electricity to Cumberland, Springfield and Romney, and points in West Virginia. F. Ernest Brackett, of Cumberland, is engineer.

GAS POWER APPARATUS

Producers, Engines and Industrial Applications

IGNITION POINT OF INTERNAL COMBUSTION ENGINES

Effect on Efficiency

By Reginald Trautschold

THE flywheel plays an even more important part in regulating the useful power output of an internal combustion engine than it does in the case of a steam engine, and it is largely for this reason that the ignition point in an internal combustion engine is of paramount importance. In an internal combustion engine, a power stroke occurs once each revolution of the flywheel, in a 2-cycle engine, while in the more generally efficient 4-cycle engine, the power impulse occurs only every second revolution.

In a steam engine, impulses are almost invariably given twice during each revolution, so that the kinetic energy that is stored in the flywheel as momentum, during the first part of each power stroke, is called upon only to augment the turning moment during the latter part of each power stroke and to carry the engine over dead centers. The flywheel of an internal combustion engine, on the other hand, not only carries the engine over dead centers but must carry the engine load for the second half revolution of the flywheel as well as a considerable part of the first half revolution, in a 2-cycle engine, and about 2½ times as far in a 4-cycle engine—power units with one cylinder. Multiplying the number of cylinders reduces such dependence on the flywheel by a corresponding amount, of course, but in any case the regulating demands on the flywheel of an internal combustion engine are great.

It is true that of the 2 types of internal combustion engines, one in which the fuel charge is exploded (explosive type) and the other the engine in which the fuel charge is gradually ignited, or the engine in which the fuel charge burns for an appreciable part of the power stroke (the true "internal combustion" engine), the correct point of ignition is of less importance in the latter than in the former, and for this reason the explosive type of internal combustion engine will be first and more carefully considered.

There is only one correct point of ignition and this is recognized by the leading manufacturers of internal combustion engines, as shown by their failure ordinarily to provide any automatic means of changing the point of ignition. Means of adjusting the point of ignition are necessary and are provided, in order to safeguard the engine against the disarrangement of mechanisms through wear, etc., but they are only refinements and should not be made use of except when adjustments are necessary. The importance placed on this point is to emphasize the incorrectness of the practice of using a later ignition point for a rich fuel charge than for a poorer mixture, a practice that, while it has not become common, owing to the necessity of manually changing the point of ignition, is nevertheless a practice that has been suggested by certain operators and engineers. The point or time of ignition should not vary for any degree of richness of fuel charge, the governor taking care of the amount of fuel charge, etc.

Demand for power on any reciprocating engine may be considered as constant for the short interval of time from one power stroke to the next, no matter how fluctuating the engine load may be, and such constant demand for power may be represented by a rectangular area, such as rectangle CDEF of Fig. 1, which may be

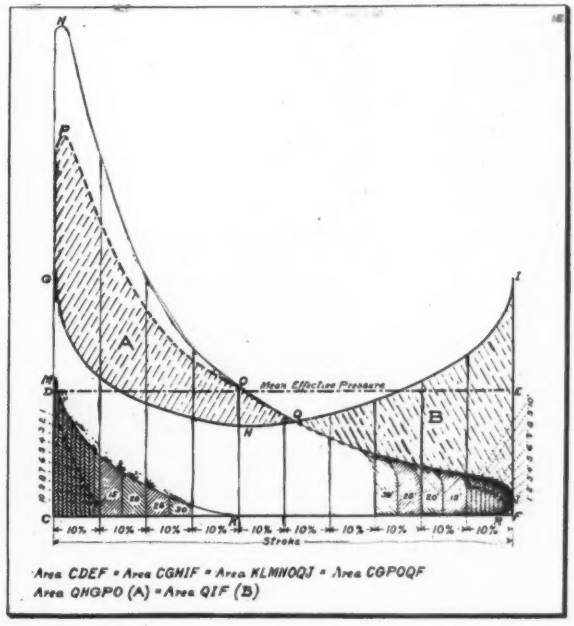


FIG. 1. THEORETICAL DIAGRAM FOR AN INTERNAL COMBUSTION ENGINE

considered to represent a constant power supply of 100 hp. per hour. The varying angularity of the connecting rod and the different positions of the main engine crank, from one dead center to the other, make the actual attainment of any such rectangular diagram of uniformity of power supply impossible, of course; hence, the necessity of the flywheel as a storage for pressure in excess of the mean effective pressure during the first and last parts of the power stroke and from which the excess power is reclaimed during the center portion of the stroke when the directly effective pressure falls below the mean effective pressure.

The varying demands for pressure depend, to a certain extent, upon the relative length of the connecting rod and the crank arm and when this relation is 5 : 1—an average of practice as far as internal combustion engines are concerned—the pressure diagram equivalent to diagram CDEF is indicated by Area CGHIF, Fig. 1.

Such pressure diagram is also unattainable by any internal combustion engine, but represents the energy required for uniform power. An indicator diagram of a 100-hp. Westinghouse gas engine—an average diagram attained from 8 cards taken with a uniform load and as nearly as possible under similar conditions of operation—drawn to the same scale as the other 2 pressure or power diagrams, is represented by Area KLMNOQJ, Fig. 1. This latter diagram is shown with the ignition taking place immediately after the crank passes the dead center and is equivalent to the Area CGPOQF.

This last area represents the average available power attainable from an efficiently designed and proportioned internal combustion engine of the explosive type. Of this

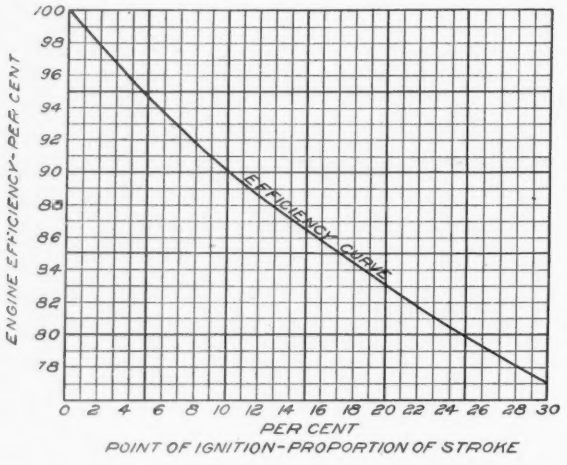


FIG. 2. PERCENTAGE OF LOSS IN ENGINE EFFICIENCY DUE TO ADVANCING THE POINT OF IGNITION

total available power, Area QHGPO (A) lies above the required pressure line GHI and is therefore in excess of the power required for uniform operation of the engine. At point Q in the power stroke, at about 52.5 per cent of the forward travel of the piston, the pressure exerted on the piston and that required for uniform generation of power are equal, but beyond this point the thrust on the piston falls below the pressure required, the lack of thrust being represented by Area QIF (B), and this amount must be reclaimed from the stored energy of the flywheel, in order to obtain a uniform turning moment.

Area QHGPO (A) and Area QIF (B) are equal, therefore, the former representing the excess energy stored in the flywheel during the first part of the power stroke and which is reclaimed, represented by the latter area, during the balance of the power stroke. In reality, this total amount of useful energy is uniformly distributed over both a forward and return stroke of the piston in a 2-cycle engine and over 2 forward and 2 return strokes in a 4-cycle engine. This distribution of energy over several strokes of the piston makes no difference as far as advisable concentration of power is concerned, however; so maximum efficiency, as indicated by the diagram, will indicate the best practice for any type of explosive engine, and any losses that may become apparent on Fig. 1 from varying the point of ignition will be correspondingly the same for either a 2 or a 4-cycle engine.

Should the ignition of the fuel charge take place before the dead center point was passed a "backfire," or impulse against the direction of engine rotation, would, of course, take place and may be disregarded in this discussion, therefore, being evidently disastrous to the efficiency of the engine. Advancing the point of ignition one per cent of the stroke would not decrease the work of compressing the cylinder contents, fuel charge, but would entail a certain unnecessary amount of work of compression. In other words, Area (A) and Area (B) would both be reduced. There would also be a slight drop in maximum pressure against the piston, owing to the greater cylinder contents in which the exploded charge could expand at the instant of ignition; but the thrust on the piston at the point of stroke where ignition took place would be the same, for all practical purposes, as the thrust at that point when ignition took place immediately on passing the dead center. During the first 1 per cent of the forward stroke of the piston there would be no thrust on the piston from the ignited fuel charge and hence, the power output of the engine would be less than in the former case by an amount equal to this loss and the work of unnecessary compression.

The loss due to unnecessary compression is represented by shaded area 1; the second loss, though not so apparent on the diagram, is, to all intents and purposes, equal to shaded area 1′, at the end of the power stroke. Similarly, advancing the point of ignition 2 per cent would entail the loss of shaded areas 1 and 2 and 1′ and 2′ from the diagram representing the required power for uniform operation of engine, and reduce the power output by an equivalent amount. A point of ignition at 3 per cent stroke would represent a loss in efficiency equivalent to shaded areas 1, 2 and 3 and 1′, 2′ and 3′, and so forth and so on.

Figure 2 indicates graphically the percentage of loss in engine efficiency, explosive type of internal combustion engine, due to advancing the point of ignition from its correct location, immediately after the commencement of the power stroke, various amounts up to a point at 30 per cent of the power stroke, the efficiency curve being plotted with points of efficiency corresponding to points of ignition.

In internal combustion engines of the true "internal combustion" type, those in which the fuel charge burns for an appreciable part of the stroke, the losses in efficiency due to belated ignition, though approximately equal in amount for engines having cylinders of the same size to similar losses in explosive types of engines, do not reduce the efficiency of the engine to quite the same extent. This is due to the more uniform distribution of thrust in such engines; that is, the distribution of pressure on the piston is more nearly like that in the ordinary steam engine, and, therefore, the ratio of mean effective pressure to the length of stroke is less than in the case of the explosive type of internal combustion engine. For instance, in engines of the Diesel type, where compression is carried to a high degree, a portion of the stroke immediately following injection of fuel may have nearly uniform pressure, though usually a gradual drop in pressure takes place in such engines while combustion is still occurring and the piston advances.

In other true "internal combustion" engines, where compression is not so great, the pressure on the piston, between the points of first injection of fuel and the cessation of fuel supply, may even increase to a maximum

at a point appreciably later than when the fuel is first forced into the combustion chamber and cylinder. Delay in injecting fuel into these engines not only entails the losses of compression, etc., noted in the case of the explosive type of engine, but may even cause failure of fuel ignition by reduction of temperature of the compressed air charge through its expansion. Of course such failure to ignite the fuel charge is not likely to occur, owing to the heat retained in the engine cylinder once the engine has actually started to operate.

STARTING VERTICAL GAS ENGINES

Precautions in Setting and Testing Governor and Details of Adjustment of Spark and Mixture

By John S. Leese

AFTER the erection of the engine has been completed, the crank case should be thoroughly cleaned out, and oil may then be put in up to ½ in. above the strainers which are usually provided for straining the oil used in the forced feed lubricating system.

Sight feed lubricators should also be filled, any pipes leading from them should be disconnected as nearly as possible to the point at which they deliver the oil, the pipes tried, to see that enough oil is coming from them. The pipes should then be coupled up again and oil run through them so that it will get to the part it is intended to lubricate. After starting up particular attention should be paid to all the connections in these pipes, as leaking joints may involve serious trouble.

If the governor is of the centrifugal type the governor springs should be uncoupled from the weights and the gear worked by hand, opening and closing the governor valve in order to see that the latter is working freely. The valve should then be set to shut off at the no-load position, that is to say, when the weights are at their widest distance apart. The governor springs may then be replaced and adjustments made so that the governor will completely close the valve at no-load speed. This is a precaution to ensure that the engine will be under the complete control of the governor when it is started up for the first time.

At the first starting up the engine will probably "hunt," due to the valve being completely closed at no-load speed, but this "hunting" can easily be remedied by after adjustment. On the majority of engines this may be done while the engine is running. The gas and air valve should be adjusted for full load while the engine speed is being set in this manner.

Next oil all moving and sliding parts, such as inlet valve, spindles and guides, rocking levers, push rods, etc., and fill up any lubricators and greasers. After this has been done the engine should be barred round by hand to make sure that nothing is touching and that all is clear inside the crank case.

Setting of the ignition should be checked over to see that the spark, when fully advanced, takes place about 25 to 30 deg. before the dead center. The angle may be conveniently checked by an adjustable spirit level.

When the engine is still being barred round it should be noticed that there is enough clearance between all levers, valve-rods and the valve stems so that all the valves will close tight. Valves held from their seats will result in pitting and corrosion and also in leakage of compression and explosion pressures. When the barring round is finished the engine should be left in the starting position, that is to say, 35 to 40 deg. after the firing point on any of the cylinders which are fitted with air starting gear.

Next, all the necessary main valves, for instance the exhaust, gas supply to engine stop valve, and air supply to engine stop valve, should be opened. The circulating water valve should be opened immediately after the engine has started. The bypass valve on the pressure gage connected to forced lubrication systems should be practically throttled off. This is necessary so that the oil will get to the bearings in the shortest possible time on the first starting up. After running for a little while the valve may be adjusted so that the correct amount of pressure is registered on the gage.

The gas cock in the engine manifold can now be opened, the gas and air mixing valve adjusted, the ignition switched on, and the engine started by suddenly turning on the compressed air. The engine should start firing after making 4 or 5 revolutions. As soon as the engine gets away on the gas the compressed air must be shut off and the oil pressure gage watched so that the pressure can be reduced to its correct amount before the gage pointer reaches the end of its scale. If this precaution is not taken some damage may be done to the gage and possibly to the piping system. The bypass should be adjusted so that a steady pressure is recorded on the oil system.

The engine should now be allowed to run for 15 min. or so, while the attendant adjusts the water supply to each cylinder by closing the valve serving each cylinder and opening it for 3 turns. By this means an equal quantity of water will be supplied to each cylinder. Should any one cylinder appear to heat up more than the others the water supply for this cylinder may be increased and the valve adjusted until all are at an equal temperature. During this run the speed of the engine should also be checked and with a full load mixture on the mixing valve all "hunting" should be eliminated.

If it is necessary, tension of the main governor springs may be altered, or this may be done later. After, say ½ hour run, provided there is, in the mean time, no appearance of undue heating, the engine should be shut down and all the inspection covers removed from the crank-case for an examination of the bearings, rod-ends and camshafts. If everything is o.k., the doors may be put back again ready for further running. In this connection it will be noticed that on no account must a naked light be taken into the crank case. Serious explosions with injuries to the inspector have often occurred through neglecting this precaution.

The main governor springs should be the correct tension to account for 95 per cent of the speed, and at this speed there should be practically no tension on the auxiliary speeder spring. If the speeder springs are used for more adjustment than this, coarse governing will follow.

The engine may be started up again and run for a quarter of an hour or so at full speed. If at the end of this period the bearings are still cool, full load may be put on the engine.

It often happens in new engines that when they are called upon to run light for considerable periods, the spark plugs become wet with oil and consequently miss fire. This is due to the new rings not being properly bedded into the cylinders, thus allowing oil to be forced up into the combustion chamber. Before going on load, therefore, it is advisable to examine the spark plugs so that there will be no miss-firing on the light load.

Another frequent cause of miss-firing at light loads with quantity governing is insufficient compression. By opening the indicator cocks it will be easy to determine

which cylinder is miss-firing. The remedy is to adjust the governor mixing valve so that it takes more gas at high speed and to adjust the supply of mixture by the plug cocks.

Best results are procured by allowing the jacket water to leave the cylinders about 130 deg. F., or at a temperature as hot as one can bear on his hand.

After having had the engine running on full load for some hours, and after the crank case and parts have become fairly heated up, it may be found that the exhaust valves are not seating. This is due to the expansion of the spindles reducing the clearance between the rockers or tappet rods and the valve stems. In cases where the valve rockers are mounted on the cylinders, the expansion of the latter will increase this clearance instead of decreasing it. This will mean insufficient opening of the valve. The correct clearances for the inlet and exhaust valves vary with the size of the engine, but the manufacturers will invariably supply these figures on request, and they should be checked when the engine is cold.

For constant quality variable quantity governing the following mixtures are approximately correct: Town gas, 500 B.t.u. per cu. ft., 8 volumes of air to one of gas; producer gas, 135 B.t.u. per cu. ft., 2 volumes of air to one of gas; coke oven gas, 400 B.t.u. per cu. ft., 5 volumes of air to one of gas; blast furnace gas, 90 B.t.u. per cu. ft., 1.5 volumes of air to one of gas.—The Isolated Plant.

FURNACE DETAILS FOR SMOKELESS COMBUSTION

TO secure suppression of smoke, the Smoke Department of the City of Cleveland has issued a pamphlet giving requirements, which are to be considered only as approximate minimum limitations. Many boilers make smoke and are inefficient because of the short travel of gases before striking the cold surface of the boiler; lack of combustion chamber; lack of draft, often caused by inadequate stack, constricted breeching or damper, or both. The following rules are for hand-fired, horizontal tubular boilers burning coal.

Height between dead plate and shell for shell diameter, 36 in., 30 in.; 42 in., 32 in.; 48 in., 34 in.; 54 to 72 in., 36 in.; 78 to 84 in., 42 in.

Suitable firebrick construction must be built above and behind bridge wall, followed by a deflection arch of firebrick. The thickness of the first grade firebrick shall not be less than:—9 in. over firing doors on inner side; 9 in. of (arched) firebrick over the firing door liners; 9 in. for side walls adjacent to grates up to height of bottom line of shell back as far as the rear edge of the bridge wall; 4½ in. for side walls above the height of the bottom line of the shell from front to end of arch system, headers every fifth course; also side walls between rear end of bridge wall and 12 in. back of rear arch, same construction.

Side walls from combustion chamber floor throughout setting behind arch system, a good second grade brick 4½ in. thick, with headers every fifth course; bridge walls, front and back faces and top, to be not less than 9 in. of first grade firebrick. All arch spans of more than 48 in. to be of 9-in. wedge brick, no double arch of 4½ in. to be of 9-in. wedge brick, no double arch of 4¼ in. being substituted.

Brick over door openings in combustion chamber to be arched, and combustion chamber to be paved with firebrick on edge, which may be second grade. A protecting pier to be in front of blowoff pipe, and back wall and back arch construction to be of first grade firebrick. The front girth seam must be protected by firebrick covering, with deflection arch under courses to be of wedge or bullhead brick.

Sidewalls must be not less than 20 in. for shells up to 54 in. diameter, and 24 in. for shells 60 in. and upward in diameter, and be built solid.

Vertical stays are to be placed with reference to arch construction, with rods extending from outside wall to outside wall, and lower rods placed below the combustion chamber floor. Longitudinal stay rods also are to extend from outside to outside. Metal reinforcement to take up arch thrust must be exposed on outside of setting walls.

Herringbone or Tupper grates are not permitted.

For steam pressure of 60 lb. or greater, steam jets, either operated by hand or automatic, the latter advised, are to be used, entering the front of the furnace, the minimum number for boilers up to 48 in. diameter being 2; boilers above 48 in. to 66 in. inclusive, 3; boilers above 66 in., 4.

Firing doors are to have panels with openings aggregating 4 sq. in. per square foot of grate and doors to control such openings, the operation of the doors preferably to be automatic.

Area of uptake above tubes and all sections before entering stack are to be at least 25 per cent in excess of the tube area which it serves, or ¼ of the grate area, the minimum section never to be less than ½ the section entering the chimney.

The breeching should be as straight as possible, and where turns are unavoidable, to be made with as long a sweep as possible. Drops in the breeching are to be avoided.

A damper must be provided for each boiler, which, if placed in the uptake, is to have effective opening at least 25 per cent in excess of the tube area, or ¼ of the grate area. Also place a damper in breeching controlling all boilers.

Opening in the extension sheet of the boiler for connection to the breeching to be at least 25 per cent in excess of the tube area, or ¼ of the grate area.

A round or square breeching is preferred to one which is wide and shallow; a breeching of a section in which one dimension is over twice the other should never be used, and long exposed breechings should be covered.

Area of the chimney is to be 20 per cent in excess of the combined tube area, or 1/5 the grate area. For stacks of exceptional height, a somewhat less area may be approved.

ETERNAL VIGILANCE is the price of efficiency. Every man must be his own watchdog, and his first duty will be to bark at himself. A lot of us would like efficiency if it did not interfere with our own ways. How angry was that officer to whom a friend reported that the improvement in his factory methods which he sought must begin by radical changes in his own behavior.

It takes a strong man to be highly efficient, because he must win a victory over himself before he can pass the threshold of the temple of efficiency. He must learn his own ignorance and then put it away. He must abandon prejudices in mind and habits in action, must throw away unbeliefs, learn to think that little which once seemed large, and that vital which was deemed unimportant. It will be clear that this ideal of efficiency is something different from merely improving a method here and there, getting a new machine, or hiring an abler superintendent. These may be done without grasping the first principles of efficiency.—Hon. W. C. Redfield, U. S. Secretary of Commerce.

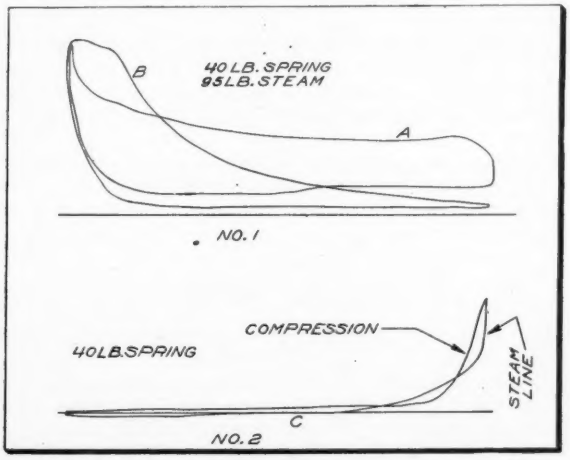
LETTERS DIRECT FROM THE PLANT

Cash Paid For Ideas Accepted. Sketches Desirable.
We Make The Drawings

BOUNCING DASHPOTS AND RACING ENGINES

WHERE there are bouncing dashpots, racing engines are sure to follow, when main belt breaks or load is very light. Numerous cases of engine and flywheel wrecks, where the governor and safety arrangements, and also the valves have shown no effect in their setting, have puzzled the experts as to why the engine should run away, and in none of the verdicts have I ever heard of the dashpots being considered as a cause of the trouble.

Now there are no engineers with any variety of experience who have not had this bouncing to contend with, either in vacuum or dead weight dashpots; but my experience has been that the vacuum dashpot is the more troublesome. It is self evident that the governor has no power over this defect, although it may be in perfect shape, as may also be the valve setting. The great cause I have always found is entrap-



CARDS FROM RICE & SARGENT ENGINE. A, WHEN DASHPOT WAS BOUNCING. B, WHEN NORMAL. C, SAFETY VALVE STOP OUT, GOVERNOR DOWN, STEAM VALVES NOT PICKING UP

ped air getting under the plunger and not space enough around the plunger to let it escape so that the plunger can find its seat and close the steam valve.

In the case of a vacuum dashpot, one would say that if there were space around the plunger, it would destroy the vacuum; and so it would if there were much space, but if there is 1/32 in. space, it does not materially affect the vacuum. and gives the air which is bound to get in a chance to get out, because the dashpot is seated in about 4 times the length of time that it is being lifted, hence the valve is lifted so quickly that 1/64 in. space won't do much damage. It is always in the case of short cutoff, where the valve is only lifted slightly, that the trouble comes.

Take, for instance, a double eccentric Corliss engine of any standard make. Now it is a well-known fact that with double eccentric, where it is desired to get a ¾ cutoff, that the steam valves can be lapped only enough to close the port "when the dashpot is hard down" or very nearly line and line. Now suppose the dashpot accumulates some air that does not readily escape and holds up ⅛ in., it will be obvious that if the steam valves were lapped 1/16 in. with dashpot hard down, that it will now hold the valve open 1/16 in., and this is especially true when the governor is up in its shortest cutoff as, for instance, when the load is off.

Now what would 1/16 in. opening of steam port do if the main belt should break? There is nothing to prevent a wreck no matter how well the safeties are set or the valves squared.

I once took charge of a 400-hp. double eccentric Corliss engine which was in excellent shape except the bouncing dashpot, which had been the cause of much trouble to those who had preceded me. It was found that with light loads, as above stated, the dashpot did not lift high enough to get a good vacuum to pull the plunger to its seat and expel the air that had been drawn in, but held up ⅛ in., which admitted steam to the cylinder, and although the governor went up to its limit, it was kept from running away only by the friction of the line shafting which just about used up the amount the valve was held open. Now, after dismantling the dashpot, nothing could be found that would have caused any trouble, so I stopped putting oil on the plunger and I never had any trouble with it except once, when I forgot and put oil on the plunger and immediately my trouble began, but ended as soon as I wiped it off. It was this film of oil around the dashpot that would not let the air out, and this had been the trouble with this engine for years.

With single eccentric engines, this defect is not so pronounced, because, where the valves have the usual amount of lap of ⅛ to ½ in., when the valves are hooked up and wristplate central, it would have around ¾ in. lap when dashpot is seated, hence the valve would have to hold up much more in this case to admit steam to the engine.

I had a similar trouble in another plant, and completely eliminated the trouble by putting on a ⅛-in. check valve piped to the body of the dashpot ¼ in. from the bottom, so that in coming down any air was expelled through the check valve, and after the plunger passed by the check valve pipe hole, a partial cushion was formed. By this method it is not possible to get air enough to open the steam valve. It is always well to note the amount of clearance between the catch block, because if the clearance is excessive, it means that the dashpot rod is too long and valve is nearer open than if shortened and holding up will give steam, where it would not if catch block had 1/16-in clearance.

A. C. Waldron.

THE WATER COLUMN

IN regard to the article, "The Water Column," by Receiver, in some ways I agree with him and in some I don't.

In regard to the valves between column and boiler, I believe they should be there and kept in condition to be of some use when you want them and want them badly, though I have seen a boiler properly roasted because they were shut when the boiler was washed and not opened when it was closed up. The man on shift when the boiler was washed had not filled it as it was almost time for him to go off. The other man saw water in the glass and put a fire under the boiler, and as it was one of a string of 5, didn't pay any attention to it for the next half hour or so.

When he looked to see if he had any steam yet, the boiler seemed to be mighty hot but no steam showing, he tried to blow the column down and it wouldn't blow, then he discovered that he had no water and didn't waste any time shutting off the gas. The result was we had to roll all the flues in both ends, which was getting off pretty lucky.

As to the columns themselves, it depends a great deal on what kind of water you are using. With water such as you get on the deserts around the oil fields these so-called safety columns are anything but safe. With bad water, I don't want any floats, levers, and other miscellaneous contraptions inside the column and pipes. You'll have enough stuff in there to keep you guessing whether it is telling you the truth or not, without putting it there.

To give an idea of what the water is like, in one place we carried 150 lb. steam and 100 deg. superheat, and anywhere there was the least steam leak, as on a valve stem, there would be icicles, as I call them, hanging down 6 or 8 in. long, if you did not break them off and tighten up the stuffing box. I blew the columns down every half hour and then have found them so badly plugged that it would be necessary to take a hammer and tap on the pipe and valve to loosen the mud and scale so that 150 lb. could blow it through a 1-in. straight pipe. And when boilers pulled over, which they would do quite often if you were pulling a heavy load, the water didn't just wiggle in the glass, there wasn't any in it, except a few drops chasing each other up one side and down the other. And when a glass blew out, well, there just wasn't any glass.

Also I don't want any automatic shutoff on the glass connections. I'll take my chances shutting it off with the chain or hand wheel when necessary. If a glass goes out I want it to tell me about it and keep on telling me till I come and shut it off.

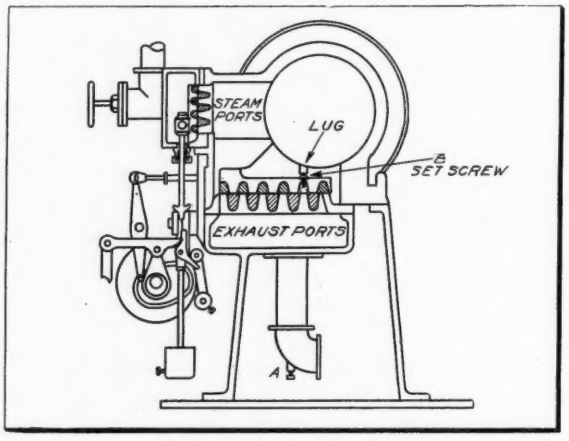
Jas. A. McBain.

DIFFICULTY WITH BROWN ENGINE VALVE

TROUBLES which engineers tell us about in steam plant operation, as recorded in Practical Engineer, are most useful and interesting and serve to keep us informed in advance as to what to expect with certain equipment.

I present this drawing to show what caused me much trouble with an old Brown engine. One day the engine slowed down and nearly stopped and upon investigation I found that the exhaust cam was not turning with the cam shaft, nor could I move it with a small bar; this caused a shutdown.

After investigating I found that the little stud B on top of the exhaust valve had unscrewed and came up so hard against the lug above it, that it jammed the valve against the seat, making it impossible to move it. I have known engineers who were running Brown engines who did not know that this stud was on their exhaust valves until so informed, and to those who do not I will say that this stud, which is really an adjustable set screw, is there to keep the exhaust valve from lifting and clattering, when the back pressure in the exhaust pipe is greater than the terminal pressure at the end of the stroke.



VALVE GEAR OF BROWN ENGINE

There is a plug A at the bottom of the exhaust pipe through which may be inserted a long socket wrench to change the position of the set screw B, but these socket wrenches are seldom found about the engines, owing to the fact that these set screws are rarely changed, due mostly to the lack of knowledge of their whereabouts. Without the wrench it becomes necessary to disconnect the exhaust pipe near the cylinder and replace the set screw. Charles Miller.

VALVE TRAVEL

AMONG the several interesting articles in the Sept. 15 issue of Practical Engineer, I was attracted by one written by A. C. Waldron, which appears on page 906. The article is clearly illustrated and shows various phases of steam distribution in a common slide valve engine. The diagrams are well worth studying.

In his Fig. 2 is shown a slide valve and its seat, with the dimensions marked thereon, the steam ports are ¾ in. wide, the steam lap at each end is marked as 9/16 in., and the travel of the valve is given as 2½ in. In looking at the diagram and its dimensions, I applied a simple rule to it that I had learned, relative to the valve travel. I do not get exactly the same figures that Mr. Waldron gets, but I believe the difference can easily be explained. His diagram gives 2½ in. for the valve travel, which no doubt is correct. But by rule, it might at first be thought to be 2⅝ in. travel; not much difference, I grant, yet to the thoughtful inquirer enough to suggest the question, why any difference at all?

The rule to which I refer is this:—To find the travel of an ordinary slide valve, add together the width of the 2 steam ports, and the sum of the steam

laps at both ends of the valves. Applying this to valve in Mr. Waldron's diagram, we have: ¾ + ¾ + 9/16 + 9/16 = 2⅝ in. travel.

This travel will give a full port opening at both ends, so it would seem that in Mr. Waldron's case the steam ports are never fully opened with the 2½-in. valve travel. It is nothing unusual to find engines with slide valves that do not fully uncover the steam ports. In such cases the width of the steam ports virtually becomes the amount the valve uncovers them and the rule quoted still holds good.

Charles J. Mason.

A PECULIAR ENGINE BREAK

IN my young engineering days I took charge of a big slide-valve engine, old type. I had run only a few days when the break occurred. The governor was not in use, we had to throttle the engine by hand so there was no time to nap and fire the boilers. They were an ancient type of the Gardner family.

The engine was running all right when there came a report like a gun and the engine never finished the revolution. I ran up and shut the steam off. At first my eye saw no cause, but second sight revealed the trouble. There was one rocker arm stand broken

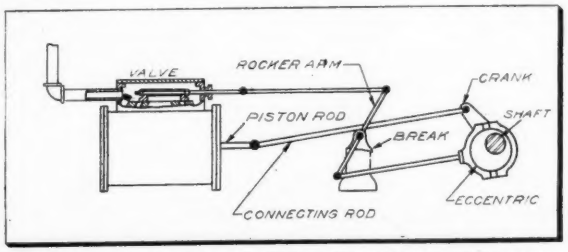


DIAGRAM OF VALVE GEAR SHOWING LOCATION OF BREAK

and pulled over out of plumb. We removed the chest cover and took a look at the slide valve. What we saw was a wonder. Part of the governor valve had come down with the steam and as the steam pipe came in the side of the chest, there was trouble when the valve met this obstruction on an unfinished travel; something had to give. The blacksmith came to the rescue and strapped the broken castings and that broken part never gave any trouble.

But the eccentric always showed that sudden shock by not running true for years after the trouble was past. It did not ruin the valve, only dented the brass somewhat.

Before long we put the old governor in use. It worked well, but in time was cast aside for a new sensitive governor.

G. H. Brough.

SOAP ON BEARINGS

W. S.'s experience with a hot bearing, as told in Aug. 15 issue, puts me in mind of some experiences I had while engineer in a large paper mill. A 6-in. shaft that carried the cones for speeding up purposes ran hot one night; the oiler tried all different kinds of dope that he had in stock. We finally took the cap off the bearing and covered the shaft with graphite and grease; it did not seem to help matters any.

The box was near the washing machine, that we used to wash our overalls, and as we had given up and were about to shut down, the oiler picked up a cake of naphtha soap that we used in washing and held it on the shaft; the result was surprising, it stopped smoking before it had made 25 revolutions and with a little soap once in a while and oil and graphite between, we had it cool, cap back on, and running normally inside of an hour.

The oiler told me a short time after that, of cooling an 8-in. beater shaft in the same way. As W. S. says, it is generally negligence on somebody's part that is the cause of a bearing warming up, and when once started it is a delicate job to cool them without shutting down.

V. C. Wood.

USE AND ABUSE OF REFERENCE MARKS

THE editorial under the above caption is of so much interest that I cannot refrain from discussing it. A great many of the older machines have no reference marks at all, except on the ends of valve stems of the Corliss type, and when necessary to make adjustments or repairs it requires a great deal of time and trouble to get the parts re-adjusted as they were before. In the case of a piston rod screwed into the crosshead, and marked with a prick punch, the mark is unreliable. The usual way of unscrewing the rod is by the use of a Stillson wrench, as no other wrench can be used. There is, however, a better way and that is to use a bar or a short piece of pipe and a rope or chain. Fasten one end of the rope or chain to the bar, and wrap it 3 or 4 times around the rod holding the end tight. This will usually turn the rod after the check nut is loosened. This method will not score the rod or obliterate the marks. The valve stems and eccentric rods are usually made with a flat spot on which a wrench may be used, but if they are not made this way the same means may be used to turn them only using a smaller bar and rope.

In regard to placing reference marks, the right way is to use a standard distance for all rods. For instance the balanced valve, and 4-valve engines which I am running have all reference marks exactly 3 in. apart. This applies to piston rods, valve gears, eccentric rods, etc. That method causes no confusion and is easily remembered. The points might be 4 or 5 in. apart as long as all are the same. On a large engine it might be better to have 2 standards, say 3 in. for the valve rods, and 6 in. for the piston rod and eccentric rods. In my case the valve gear has a center punch mark on the valve rod head about ½ in. from the end. After the engine was erected, and the valves properly set, a similar mark was placed on the rod exactly 3 in. from the first. The rods are marked on both ends. The marks are set in an oblong circle similar to the letter "O" which makes it easy to locate. If for any reason the valve gear is changed the rods are drawfiled and polished to remove the old marks, and new ones are made in the proper place. By this method there can be no confusion as to the proper marks. The piston rods are marked in the same manner.

On the 4-valve engines the cranks on the end of the valve stems are held by set screws, and also the rocker arms, and there is a chisel mark on the end of the shaft, and one on the hub of the crank to correspond to it. These are never changed, but if they should be taken off or should slip they can be quickly reset. The eccentrics and shaft also have corresponding chisel marks to show the proper position. The striking points of the piston should be located on the edge of the guide opposite the center of the crosshead and also marks showing the limit of travel are placed on the guides; the difference between the

two shows the clearance on each end. When the engine is turning over slowly these marks and those on the crosshead can be compared.

In screwing in the piston rod I should check up by the reference mark on the rod, and also by the striking point marks on the guide. On one large engine with a 5-in. piston rod we had considerable trouble from the piston rod screwing out of the crosshead when running. When this did happen it was only necessary to stop the engine and turn the rod back until the marks were correct, and set up the lock nut, the whole time being not over 5 min. I have seen some engines on which the wedge blocks or key and gib of the connecting rod was all scored up with scriber marks, which was confusing in keying up. I have one heavy scriber mark across the end of the rod and wedge. If I wish to make another for temporary use in keying up I use a scale and a fine lead pencil. This does not leave a permanent mark but is satisfactory for the time being. If the engine is not running properly, with this system of marking the engineer can tell in 2 min. by inspecting the marks whether the trouble is caused by disarrangement of the different parts or not.

J. C. Hawkins.

LINING UP THE ENGINE

YOUNG engineers are often in doubt as to the method of lining an engine, and have to call on a more experienced man for information. When they get instructions it is always the same method and requires much tedious and heavy work. I use a method that does away with stripping the engine. The idea is not new or original with me; but I have never seen it used except by one man, from whom I learned it.

About 25 yr. ago I was installing a large Corliss engine, when a tramp engineer came along and wanted work, and as I was a little short of help I gave him work.

When it came time to line up the engine for the final test, I was proceeding in the old way, when he told me he could show me a much better and quicker method, which he did in the following manner.

Put the engine on the head center, disconnect the rod from the crank and drop it down out of the way, and roll the crank to the back center. Take off the cylinder head, place a strip of wood tightly in the counterbore across the cylinder and with a pair of dividers find the exact center of the cylinder. Place another piece of wood between collar and boss of crankpin and find exact center of pin. Now, stretch a line A B, from 2 or 3 ft. beyond the cylinder back 2 or 3 ft. past the rim of flywheel, far enough away to clear the cylinder and all obstructions, and level with center of cylinder and center of shaft. Now take a large pair of calipers or any other tool you wish and get the line the same distance from your mark in center of cylinder and center of crankpin, C D in sketch. This is your line to work from instead of through the cylinder.

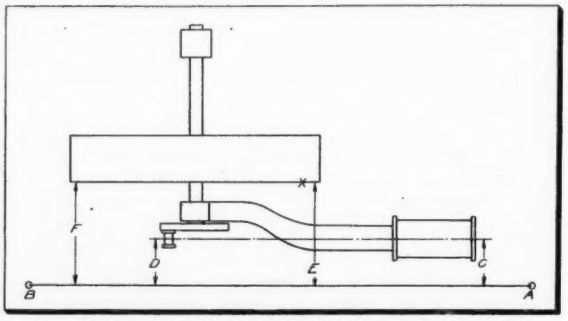
Now, take a light, stiff batten and point one end and run it through the opening opposite the guides, which you can do if you have shoved the crosshead up to the cylinder. Make a mark on your batten where the line crosses it, as at E; also make a mark on rim of flywheel, as at X, so that you will always measure from the same point. Now, roll the wheel over ½ turn and try your batten again, as at F; be sure your batten is touching the same point as before, and that it is level.

If the line crosses at the same mark your engine is in line. If not, make a mark where the line crosses, then move the outboard bearing until the line crosses just halfway between the 2 marks. If you have had to remove the outboard bearing far, it is best to try your measurements again from center of cylinder and center of crankpin to see if the movement threw it out any.

I consider this method much better where you are lining a Corliss or other heavy engine, where the cylinder is rigid with the main bed, and the guides are either bored or planed true with the cylinder.

It is more accurate from the fact that you have a longer space to work from, as from the front cylinder head to the full throw of the crank as against the length of the cylinder, and the full diameter of the flywheel as against the throw of the crank in the old method. An error almost imperceptible in the old method might show up to almost ¼ in. on your batten in the latter method.

For leveling the shaft I prefer a plumb line to a level, from the fact that you have the full diameter of the wheel to work to instead of the short length of a level.



METHOD EMPLOYED TO LINE UP ENGINE WITHOUT DISMANTLING IT

Another thing that almost all engineers advise against, is removing the label from a gage glass with a knife or piece of metal of any kind, or using a metal rod to clean out a glass. I have been doing this for years when occasion required it, and have never had an accident that I could blame the practice for. I think all engineers "who have tried to cut a glass with a file" will admit that it takes a mighty hard piece of steel to mark a glass.

There are many things that are more destructive to a gage glass than using a knife to remove a label. A little paint, linseed oil or varnish, let dry on a gage glass will invariably break it in a short time, if there is any rapid change in the temperature, just as a piece of metal, such as a hammer, wrench or chisel placed near and left for a time.

I know there are many engineers that will not agree with me on these subjects; but it is different ideas that make this old world pleasant to live in.

S. B. Roper.

LENGTH OF INDICATOR CARDS

IN the issue of September 15, page 912, a note by John M. Leitch was rewritten, and in the course of printing, the word "cord" was transformed into "card," which changes the meaning of the item. To correct any misapprehension as to the meaning which Mr. Leitch intended to convey, his communication is here reprinted:

"It is not a mistake for indicator cards on a steam engine, Corliss type, with same spring and same cord. to

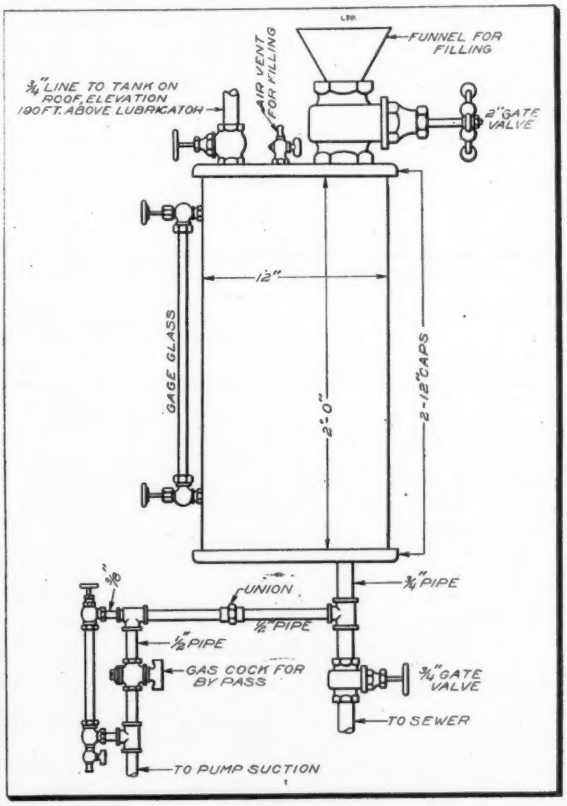
vary in length of card. (1st) Indicator cord not being properly stretched before applying to the cards; (2nd) by using too much pressure on indicator pencil; (3rd) loose connections; (4th) after all of above omitted and lost motion taken up, I never could trace any variation in length of cards; (5th) engine record not as good as some."

(We are printing this, at Mr. Leitch's request, exactly as he sent it.—Editors.)

LIQUID COMPOUND FEEDER

IN my estimation the best and most satisfactory way of feeding compound to boilers is drop by drop into the feed pump suction, the same as feeding cylinder oil to an engine. With this method the feed water is partly neutralized before entering into the boiler.

This I have ascertained by experience in feeding compound to boilers in several different ways, and the drop by drop method has given the best results. It is also economical in the use of compound as just exactly enough can be fed to the boilers to neutralize the solid matter in the feed water.



HOME-MADE COMPOUND FEEDER

The rate of feed will depend upon the strength of the compound, amount of water fed to boilers, also the quantity of solid or scale forming matter that is in the feed water which varies considerably with the locality and source of the feed water.

Following is a description and sketch of a home-made compound feeder that the writer has been using for feeding compound drop by drop into the boilers through the feed pump suction line. This lubricator is large enough for a 12-hr. run of 2 250-hp. water-tube boilers using lake water for feed.

It is made of a piece of 12-in. pipe 2 ft. long with 12-in. pipe caps screwed on both ends. The upper cap is tapped for 2 ¾-in. pipe thread and 1 2-in. pipe thread. The ¾-in. pipe and valve on the left are connected with the cold water house tank on the roof which has an elevation of 190 ft.; this is to give a head or pressure on top of the compound to insure a good steady full feed to the needle or feed valve on the sight feed glass. The other ¾-in. connection is for an air vent to be opened when filling the lubricator.

The 2-in. connection and gate valve is for filling the lubricator. On the bottom cap is the sight feed glass and connections from an old hydrostatic lubricator. Below the tee going to the sight feed connections is a ¾-in. gate valve with a pipe leading to the sewer; this is used for flushing out or cleaning out the sediment that sometimes settles in the bottom of the lubricator.

The lubricator can be secured to the wall or any other convenient place by means of straps, or it can be set on a bracket or stand if so desired.

C. E. Anderson.

ABOUT CRUDE OIL BURNING

IN the issue of August 15, I note that Mr. Hawkins differs from my statement in regard to the amount of crude oil necessary to do the work in a plant, and that he quotes from eminent authorities whom he says may be considered accurate. I believe that anyone with practical experience in the oil burning business will quickly tell whether those eminent authorities or myself are right on this question.

Of course, the kind of burner, type of furnace, and the boiler details are to be taken into consideration and after many hard dollars went into the coffers of the railroad companies for traveling expenses I decided that the Higgins burner was the best. I visited that Galveston plant and what I know about it caused me considerable surprise. Also, as to the plant at San Antonio, I have been there several times and hung around there quite a bit and that plant comes nearer to saving money by being economical than the most of them. In nearly all economical plants, they are so because they do it on paper. As an instance, a certain college in Oklahoma put some fine figures in the public eye; yet, when the manager of my firm and myself went to that place we found that the most of the tests were paper tests, and the same was true in Texas. I wish that Mr. Hawkins could come to the country where I abide and go with me to the various oil burning plants here. I judge that he lives in the East and I also am a product of that section, from Cleveland, Ohio, and came west to teach the natives, how; in justice to them let me say that the East has nothing to show them and they have 5 oil burners here to one in the East.

As to the sizes of Corliss engines and speeds, I have put this question up to several manufacturers who sent me tables by which an ordinary engine is proportioned, one of which I quoted in my previous article. As to the 30 by 48 and 150 r.p.m., I find that this size at 100 lb. steam pressure and 1/5 cutoff and 70 r.p.m. would deliver 540 hp. and the engine would have a 22-ft. flywheel weighing 33,000 lb. Doubling the speed would bring this to about Mr. Hawkins' figures and horsepower, but would be running a 33,000-lb. mass at a velocity of 9900 ft. a minute rim speed and I wonder which of the boys would like to sit around and watch her go.

John Pierce.

Questions Answered and For Answer

Expert Help When In Trouble. If You Want Quick Answer Enclose a Stamp

Thickness of Steel for Tank

IF we required a tank of 20 ft. in diameter and 20 ft. high, of steel plate of ordinary thickness, with longitudinal seams, single riveted, lap-joint; find the thickness of the plate, allowing ⅛ in. more for corrosion.

N. N. B.

A. The question in regard to the tank evidently is meant to apply to a tank for the holding of water. In this case, the pressure would be determined by the height of the tank, 20 ft. Each foot in height gives a pressure of 0.434 lb., so that the pressure at the bottom would be 8.68 lb. per square inch.

We have, then, the regular rule for the thickness of a circular shell under pressure:—The thickness is equal to pressure times the radius in inches divided by (the safe tensile strength of the material times the efficiency of the joint).

The pressure is 8.7 lb. practically; the radius for a 20-ft. tank, 120 in.; the strength of steel is to be taken as a breaking strength of 55,000 lb. per square inch when no strength is given, and the factor of safety for such work is 5,—giving a safe working pressure of 11,000 lb. per square inch.

For a single riveted lap joint, the efficiency, with usual proportions, is 56 per cent, so that the term in parenthesis becomes 11,000 times 0.56. Carrying out the computation, gives 0.1695 as the thickness in inches.

From your problem, ⅛ in., or 0.125 is to be added to this, giving a total of 0.2945, which is approximately 5/16 in. plate.

A. L. R.

Transformer Connections

I SHOULD be pleased to have you send me a sketch showing 3 single-phase transformers with the 6 terminals of each properly connected on a 2300-volt, 3-phase primary and giving in first sketch a 3-wire 220-110-volt secondary and secondly a 220-volt 3-phase secondary. I should also like to have you illustrate how it is possible to get a 110-volt circuit from this 220-volt 3-phase secondary. Can a fourth wire be run to transformers and secure this voltage? If so, show transformer connection.

C. D.

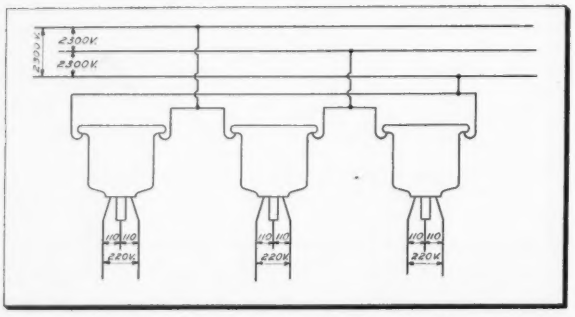


FIG. 1. TRANSFORMERS CONNECTED FOR 220-110-VOLT, 3-WIRE SECONDARIES

A. You do not state the voltage of your transformers, but from the context of your question, I assume that they are 2300 volts on high potential side. By using a Y connection, 1330-volt ($\frac{2300}{1.73}$) transformers might be used, but this is hardly a standard voltage.

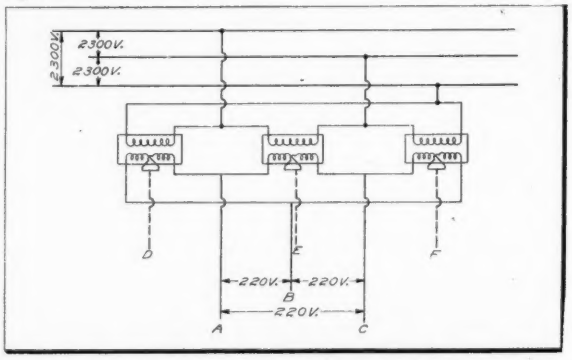


FIG. 2. CONNECTIONS FOR 220-VOLT, 3-PHASE SECONDARY

Figure 1 shows how you would obtain a 220-110-volt 3-wire system. Inspection will show that this connection is simply "single phase"—"to single phase," as the 3 single phase transformers are connected to each pair of wires exactly as though it were a single phase transmission system. This method of power distribution is common, the only precaution being that the load must be so distributed among the transformers as to maintain balanced load on the 3-phase primaries.

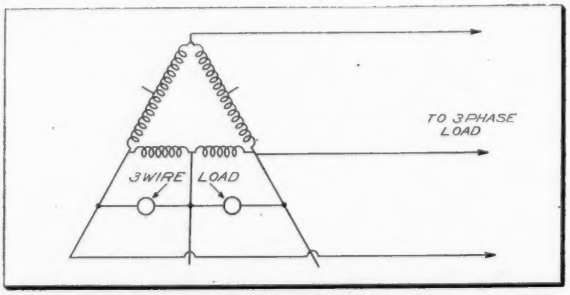


FIG. 3. TRANSFORMER CONNECTIONS FOR 3-WIRE, 3-PHASE SYSTEM

Figure 2 shows how a 220-volt, 3-phase secondary would be obtained. A, B and C are the secondary mains.

To obtain 110 volts from this arrangement, it is only necessary to tap the middle point of each or any secondary, as shown by the dotted lines D, E and F. We would obtain 3 3-wire systems—namely, A, E, C, with 110 volts from A to E and E to C; C, F, B, with 110 volts from C to F and F to B; B, D, A, with 110 volts from B to D and D to A.

Figure 3 shows a schematic arrangement of this latter plan—giving both 3-wire and 3-phase current. Only the secondaries are shown, and 3-wire load is taken from one phase only.

For further information see Practical Engineer for Sept. 1, 1913, on page 836. V. E. Johnson.

Engine Knocks

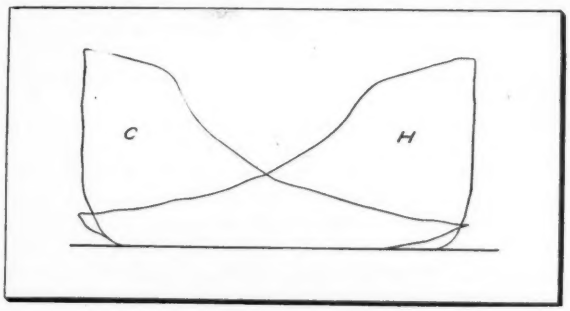
ENCLOSED please find 2 indicator cards taken from our gin engine which is a Harris Corliss. The engine is somewhat noisy and the knock seems to be in the crosshead just as it gets on center. The cards do not show full load as we have not gotten all of the gins ready and I understand the engine knocks pretty heavy under full load.

Kindly give me your criticism of diagram as the release seems to be somewhat late; the drop in steam line is owing to reducing valve in steam line.

E. C.

A. The late release as shown on the indicator cards submitted has perhaps as much to do with the noise in the crosshead as anything although it might be taken up a little more than has been your custom.

The release is much too late and too high also. A close inspection of the crank end card shows that release does not take place till the piston has started on



CARD FROM HARRIS CORLISS ENGINE

the return stroke so that there is a compression just previous to release. The terminal pressure is about 15 to 16 lb., which is wasteful.

This high terminal pressure is due to late cutoff owing to the reduction of pressure through the reducing pressure valve. Initial pressure is now only about 100 lb., while boiler pressure is 125. If there is no good reason why this reducing pressure valve should be in the line leading to the engine, open it up wide or remove it.

If these cards do not show full load, as you say that all of the machinery is not ready, you ought to arrange for boiler pressure at the throttle by putting in about 2 ½-in. bypass around the reducing pressure valve with a valve in it, or else remove the reducing valve entirely, if your period of light load is not long.

With a higher initial pressure, cutoff would take place earlier, and the steam would be expanded down to the atmospheric line, or a little above it, which would effect a little greater saving than you get now.

As to the knock in the crosshead, it may be due also to looseness of the shoes in the slide and when they are taken up it will remove some of the noise. Note if the pin is worn out of round, for if so, it ought to be turned or ground, and the boxes re-babbited or scraped to fit.

In taking up the shoes, see that the piston rod is kept central with the slide by calipering it with the engine on the crank end center. If an engine is allowed to operate for any length of time with loose bearings, it is almost impossible to get it to run quietly again.

Advance the eccentric on the shaft, say, about half an inch as a trial (as we do not know the diameter of it and therefore can not advise more definitely) and if admission takes place too early, let out a little on the steam rods (assuming that it is a single eccentric with one wristplate to drive all 4 valves) till admission takes place at about the same place as at present, or a little later. Indicate as you go along and do not try too much at a time.

G. H. Wallace.

Generator Rating

I HAVE in my care a 3-phase, self exciting, a.c. generator, and on this generator is a name plate stamped as follows: Volts 2300, amperes 25, 600 r.p.m. Will you give me a rule to figure the ampere per phase of this machine. It is not 25-ampere per phase; they tell me that it is a 40-ampere generator and I cannot understand how they get 40-ampere out of this machine. Will you please give me the true amperes and also the true watts of this generator when it has a full load, also the ampere per phase? How do you figure the power factor of the circuit? We are carrying at this time about 15 amp. on our machine, and they are adding more work on this generator every day and that is the reason why I am at a loss to know the true amperes so that I will be able to tell when this generator is loaded to its full capacity. A. H. M.

A. In the rating of alternating-current generators it is usual to place the amperes per terminal upon the name plate. Your rating, therefore, would be 25 amp. per terminal, 2300 volts, 600 r.p.m.

To figure the capacity of this machine, multiply the amperes per terminal by the potential difference between terminals and by 1.732. In your case, this becomes 25 times 2300 times 1.732, equals practically 100,000 watts, or 100 kw.

You can determine the power factor of your circuit by the use of a voltmeter, an ammeter and a wattmeter,—putting these instruments on one phase of the line and taking instantaneous readings. If you divide the wattmeter reading by the product of the volts and amperes, the result will be the power factor of the circuit. This method is the most practical, unless you have a power factor meter. R. E. T.

If Your Questions Are Not Answered

AS several correspondents have sent in questions which had no sufficient address, or were not signed, it seems necessary to call attention to the fact that no attention is paid to any communication which does not give the full name and address of the writer. This is not used for publication but as a guarantee of good faith and so that, if necessary, further information may be asked in order to answer questions satisfactorily. Unsigned letters are carefully deposited in the waste-basket.

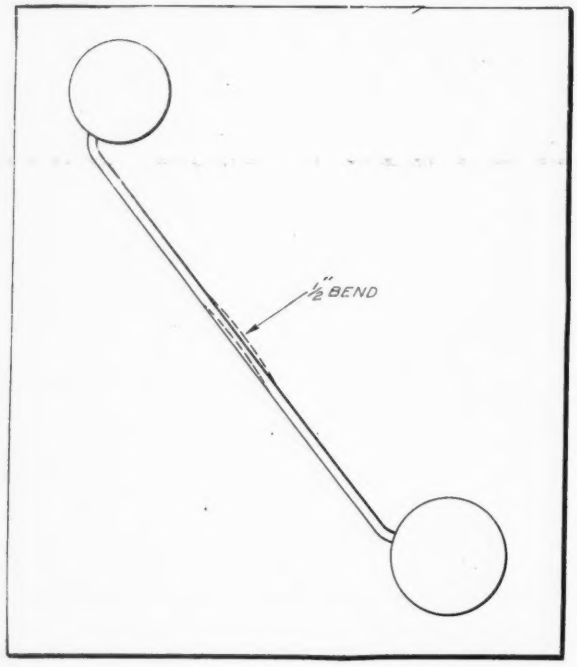
THE MINNEAPOLIS ASSOCIATION of Stationary Engineers at its recent meeting installed State President Mueller, who was absent in Germany at the time of election to that office. They also received a silk altar flag from the National Association for large increase in membership during the past year, and discussed plans for the coming banquet and booster meeting, with campaign for new members.

Problems For Discussion By Readers

What Would You Do If You Had These Conditions To Meet?

Bend in Tube of Stirling Boiler

I WISH to know what may have caused the front bank of tubes in a Stirling boiler to bend upwards about ½ in. This boiler works at 100 lb. pressure, feed water is the returns from a heating system, and the boiler is not worked hard. The setting and foundation seem all right, and the drums are level. The sketch indicates the location of the bend. S. T.



LOCATION OF BEND IN BOILER TUBE

(There is a phenomenon presented here which readers are invited to discuss. The point of contention is why the tube bends up instead of down.—Editors.)

Telephone Trouble

SHOULD like to hear from some of the readers in regard to the following trouble: I have charge of a power plant which sends power to a city some 30 miles away, also supplying a number of towns on the way; the voltage on the first 10 miles of line is 6600 v., on the other 20 miles, 20,000.

The company has a private telephone line which is run on the same pole line; this line is transposed every 7 poles and on the last 20 miles it is on separate arms some 6 or 8 ft. below the power line.

There are 10 2500-ohm instruments on the line all having arresters; now, at times it is impossible to use the line at all. At any time it is noisy, and one is liable to get a bad shock if not careful to keep the hands off the metal parts of the phone.

H. B. P.

Length of Belt Required

IN the Sept. 1 issue, W. L. B. asks how much he should shorten his belt to have it carry the load. He does not state how many horsepower the belt is required to carry, and it is possible that the reason that it slips with 13 in. taken out is that it is overloaded. It is more than 13 in. short because the pulleys must be considered as 12 ft. plus ½ the thickness of the belt, and 68 in. plus ½ the thickness of the belt. That would measure more than the 82 ft. 3½ in., therefore the belt is stretched more than 13 in. As the belt has been used several years, it will not stretch much more, and can be made a moderately loose fit and still carry its load.

It is impossible to give a hard and fast rule as to how much a belt should be shortened before it is placed on the pulleys. The method of tannage has a great deal to do with it. The old long time process gave a long fiber leather with more elasticity and more staying power than the short tanned leather which will not stretch as much. A prominent belt manufacturer says that "while it depends on the existing conditions as to how much shorter the belt should be cut than the actual tapeline measurements around the pulleys, it is safe to say that the average leather belt in the market today should be cut 2 in. short for each 10 ft. of tapeline measurements." Thus a belt 82 ft. long should be cut about 16½ in. less than the actual measurements.

In regard to a factor for shortening belts in use the same conditions are met with as regards the different tannages of leather. That is, the short fiber leather stretches less and has a shorter life, and if shortened the same number of inches as the long fiber leather it will pull a greater load without slipping. Two per cent of its length should be the maximum that a belt will stretch unless it is overstrained. Any belt that stretches over 2 per cent of its length is liable to break if stretched to a much greater degree. This would mean that the maximum stretch of the belt in this case should not be over 19 in., and as it is 14 yr. old it would seem that 16½ in. as figured from the actual tapeline measurements would be a reasonable amount to allow and still be on the safe side. If it did not carry the load before, perhaps adding another ply, and taking out 3½ in. more will make it tight enough to do the required work. If it does not I should advise that it be run at a higher speed or a rider belt be added.

In order to determine whether his belt is overloaded, he will find the following a good transmission rule and one that leaves sufficient reserve power in the belt; divide the number of feet that the belt runs

per minute by 800 and multiply by the width of the belt in inches; the result will be the horsepower that a single belt will transmit. In other words, if a 4-in. belt runs 3200 ft. per minute according to this rule the belt will carry $3200 \div 800 \times 4 = 16$ hp. For a double belt divide the speed by 500 instead of 800 and multiply by the width as before. These are both old rules, but have been proved to be safe to use in actual practice. The speed of the engine is not stated in his case but a 12-ft. wheel running 110 r.p.m. would give a belt speed of about 4100 ft. per minute, which, divided by 500 times 21½ in., gives 176.3 hp. Calculating the safe load for the belt and comparing it with the actual load carried will show whether the belt is overloaded or not.

J. C. Hawkins.

REFERRING to W. L. B.'s article entitled "Length Required to Shorten Belt," which appeared in Sept. 1 issue, without knowing the horsepower transmitted it is not possible for us to do more than generalize. We presume the reason for putting another ply on the 2-ply, 21½-in. belt was due to an increase in the load beyond that which it had been successfully carrying for 14 yr.

As the pulleys were good size we are inclined to believe from our experience with Cling-Surface if W. L. B. had filled that belt with Cling-Surface it would not have been necessary either to shorten it or to have gone to the expense of adding another ply.

We have been using Cling-Surface for nearly 16 yr. and have found that a belt properly filled with it will transmit more of a load running slack than the same belt can possibly transmit without it when running tight.

Our suggestion therefore to W. L. B. is to treat the belt thoroughly with this dressing on both sides, at the same time keeping the belt where it is good and warm so that the Cling-Surface can penetrate where it belongs. Let him then put on the belt without taking any out and continue to apply the dressing to both back and face as often as the belt will absorb it.

The belt should run for many years before laying a knife to it. If the belt is oily it should be cleaned with gasolene before the application is begun.

Chas. F. Chase.

Increasing Capacity of Chimney

THE answer to J. W. D.'s question, Aug. 15, 1913, rests on the fact that the laws governing the flow of flue gas are substantially the same as those determining the flow of water in pipes; namely, the movement is the result of a difference in pressure between 2 points. In the case of the chimney the difference in pressure is known as the chimney draft and is always proportional to the height of the stack. Since the chimney gas is less dense on account of its high temperature than the outside air, there will be a force tending to push the light gas up through the chimney, and this force will be equal to the difference in weight between the chimney full of hot gas and an equivalent column of the cold, outside air.

For the flow of water caused by difference of pressure between 2 points:

The square of the velocity in feet per second = 2 × 32.2 × Equivalent head in feet corresponding to difference of pressure between 2 points.

Wt. of water flowing per second = Area of pipe × Density of water × the square root of (2 × 32.2 × Equivalent head in feet).

Similarly for the capacity of a chimney, neglecting flue friction, pounds of gas discharged per hour = 3600 × area in sq. ft. of chimney × K × $\sqrt{2 \times 32.2 \text{ draft}}$ × dens.

From Rankine, draft =

$$\text{Height of chimney} \times \left(\frac{7.64}{Ta} - \frac{7.95}{Tc}\right)$$

Ta = absolute temp. of outside air.

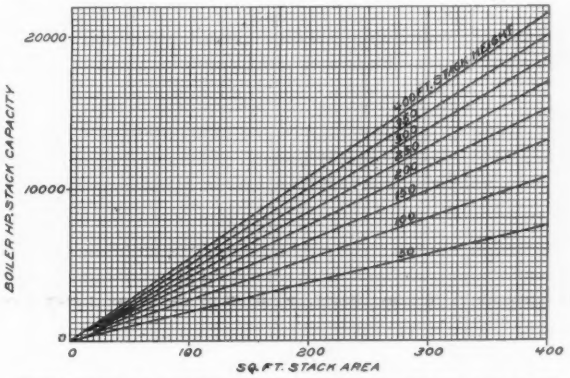
Tc = absolute temp. of flue gas.

K = a constant depending upon the physical characteristics of the chimney gas, etc.

Therefore, capacity of a chimney = $K_1 \times A \times \sqrt{H}$.

A = area in sq. ft.

H = chimney height in feet.



INFLUENCE OF HEIGHT AND AREA UPON STACK CAPACITY

It is evident that doubling the area has the same effect as quadrupling the height.

The chart shows about what would be expected of ordinary stacks with natural draft.

Haylett O'Neill.

WORK ON THE NEW SUBSTATION of the Amherst Power Co., in Chicopee, Mass., is practically completed and the plant is now supplying many of the city's largest manufacturing plants with electric power. The station is fully equipped for the present needs of the company and the equipment is so arranged that more may be added as occasion demands. Besides offering power at a much cheaper rate than that furnished by the city's municipal plant, the erection of the plant and the installation of the equipment has brought an increase of about $150,000 to the city's taxable property. It is generally believed throughout the city that the plant with its cheap power rates will serve as a magnet in drawing new industries here as there are many good factory sites available and easily accessible to the railroad.

The substation is capable of transforming the 66,000 v. which come over the company's wires from Turners Falls to 13,000 v. for consumption in the local manufacturing plants and also 600 v. for the street railway service. The ultimate capacity of the plant is 15,000 kw. or 20,000 hp., which is much more than will be needed in the city for some time.

Owing to the fact that the company has agreed to sell electricity to the municipal plant at what is believed to be a cheaper rate than the plant can manufacture the electricity itself, it is considered probable that in time the city plant will purchase all its power from the Amherst Co. and act only as a distributing station, but there is no immediate prospect of this change because of the large amount of money invested in the city plant. The company does not sell power direct to consumers, but through other companies who act as distributors.

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SAVING BY SYSTEMATIZING

The secret of having plenty of time, is to use time profitably. Recently the secretary to the president of a large corporation, said: "A young man who wants to grow fast, will study to systematize his time, and he can then accomplish more in a month than he could ordinarily in a year."

The way to start this is to take an inventory and find out where every minute of the day goes, and study the number of minutes lost during the day by needless delays, by talk which gets nowhere, and by unnecessary effort. Then study to eliminate these delays, so that your days may furnish opportunity for greater usefulness than before. Not only the time in working hours, but outside, should be scrutinized.

Few people give serious thought to the time limited in life until it is too late. At 17, the future looks like an unexplored world, with time unlimited, but J. R. Worden, of the Burroughs Adding Machine Co., calls attention to the outlook at 37, or 47, when time has acquired the boundary lines, and we look back at lost opportunities, and at long hours of well-meant but profitless labor. We begin to realize that we cannot afford to put off till tomorrow, or to miss even one opportunity.

Investigation shows that a large percentage of failures are due not to lack of ambition or hard work, but to wasting time and waiting to find out whether we are making good or not. The successes come to the man who thinks more, who systematizes, and who knows along what lines his work is successful. A plant or a business which runs smoothly, is one where the work is systematized, and the time is economized. When everything is done haphazard, no account is taken of the wasted minutes and hours, and there is much that never gets done when it should be.

WHY UGLY?

As a question of practical rather than academic interest, why should a power plant or a boiler house be ugly either in itself or its surroundings? Is a pile of cinders, ashes or boiler scale or a scrap heap of old grates and boiler tubes necessary as a front door ornamentation of a power plant? There are usually out of the way corners and waste ground enough where such accumulations may be put and if an unsightly array of coal bins seems to be an obstacle, an attractive covering of woodbine, wild hops, or wild cucumbers can be secured at comparatively little cost of money or effort. Pleasant surroundings make up half of the joy of the day's work and the public is apt to judge a man largely by the surroundings which he is willing to endure. Nature is a generous and tireless helper in attempts to beautify, and will gladly furnish a living drapery to soften and conceal harsh outlines and unlovely surfaces if her aid is invoked even half-heartedly.

STATEMENT OF OWNERSHIP, MANAGEMENT, CIRCULATION, ETC.

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Chas. Sanford Clarke, Sect'y.

Sworn to and subscribed before me this 26th day of September, 1913. Warren A. Lathrop, Notary Public.

[Seal.] (My commission expires October, 1914.)

NEWS NOTES

WORK WILL SOON BE STARTED toward erecting a brick power house at dam 28 at Huntington, W. Va.

THE ELECTRIC LIGHT CO., of Forest City, Iowa, plans the erection of a new electric light power house this fall.

PRATT, KAN., VOTED BONDS lately to the amount of $50,000 for the installation of a municipal light plant for the city.

CONTRACT FOR THE ERECTION of an electric plant to replace the one recently destroyed by fire at Clayton, Ala., has been awarded to J. B. McCrairy.

THE $800,000 STEAM POWER PLANT of the Alabama Power company in East Gadsden, Ala., will be completed soon, and the entire working force will be moved to Lock 12.

A CONCRETE ADDITION is being built on the north side of the electric light plant near the Newman Springs road at Red Bank, N. J. The addition will accommodate the switchboards and regulators.

W. R. RHOADES, 476 Broad Street, Chambersburg, Pa., is preparing plans and specifications for a 1-story brick, steel, concrete, stone and terra cotta power plant at Chambersburg for Wilson College.

PLANS ARE BEING MADE for the construction of a plant for the Marine Electric Light & Power Co., at Edwardsville, Ill., which will furnish light for the city. The plant will be controlled by former Mayor Fred Scheurer.

THE PENNSYLVANIA STATE HOSPITAL for the Insane, at Warren, Pa., will have plans prepared in the near future for a new power plant to cost about $125,000. Water-tube boilers to develop about 1600 hp. will be required.

FRANK S. PRITCHARD, formerly with the United States Graphite Co., has become associated with the Dearborn Chemical Co., at Cincinnati, as assistant to Dan Delaney, for many years manager of the Dearborn Chemical Co.'s office at Cincinnati.

A BUILDING PERMIT for the new Union Depot power house, at 401-13 West 24th St., Kansas City, Mo., has been issued to the Kansas City Terminal Railway Co. The building will be one story, of brick, stone and reenforced concrete and will cost $175,000. It will be completed by June 1. The Fogel Construction Co. will be the builders.

THE STATE BOARD OF CONTROL awarded the contract for the construction of the power plant building at the Rochester, Minn., State Hospital for the Insane to G. Schwartz & Co., of Rochester. The price is $40,420. The contracts for the equipment were let some time ago. Work will be begun as soon as possible and will be under state inspection.

THE NEW $100,000 POWER PLANT of the City Light & Power Co., of Amarillo, Tex., has been begun, and 25 men will rush the work of foundation building to rapid completion. Six hundred barrels of cement will be utilized in this construction. The building will be commodious and modern, and will be used as an exclusive engine room. It will be equipped with the best machinery obtainable. Under terms of contract the building must be completed within 60 days from the date construction began.

JOHN SCANLON, attorney of the Cohoes Company, recently received from the State Public Service Commission, of New York, a certified copy of the formal granting of the application for the construction of a hydro-electric plant in the bed of Mohawk River for the generation of electric power. The plant will be located a short distance below the Cohoes falls. A grant of land has already been received from the State. Work has already begun on a road cut through the bank to reach the site of the new electric plant. It is not likely that much can be done before spring. The company proposes to furnish power for manufacturing purposes at a reasonable rate and citizens of Cohoes confidently expect that new industries will be attracted to the city.

ROCK RIVER LIGHT AND POWER CO. has let the contract for the hydro-electric equipment of the United States' government dam at Sterling, Ill., to the Allis-Chalmers Co., of Milwaukee, Wis., for a sum slightly less than $200,000. This does not include the cost of the erection of a new power house, the excavating of rock, or the extra concrete work. When completed, the total cost, not including the cost of the concrete work already in place, will be about $230,000. The present concrete work cost about $40,000, making the total cost of installation at $270,000. The company has also let the contract for plans for the power house, which will be 208 by 52 ft.

The work of excavating for the tail race will be started as soon as possible, while concrete chutes for turbine wheels will be let by contract. Fifty or more men will be given immediate employment on the structure. Owing to the famine in men here, it will be necessary for the company to import men here to do the work. It is anticipated that the equipment will be ready for shipment from the Milwaukee shops next February, and that the installation will be complete about August 1.

Power Apparatus in Shop and Market

New Ideas In Making, Buying and Selling

ELLIOTT-PITTSBURGH TWIN HEATER

COMBINING features of the Pittsburgh feed-water heater with improvements of its own, the Elliott Co., of Pittsburgh, Pa., has brought out several new types of feed-water heaters, one of which, the twin heater, as its name implies, consists of 2 heaters in one. This feature in itself is valuable, as it permits cleaning half the heater while the other half is in service.

Figure 1 is a front view of the twin heater, which, excepting the bolts and brass fittings, is made almost exclusively of cast iron, this material having proved to resist corrosion most efficiently.

The internal arrangement of the heater will readily be understood from Fig. 2, which shows the location of the spray box and pans, also oil separator, float,

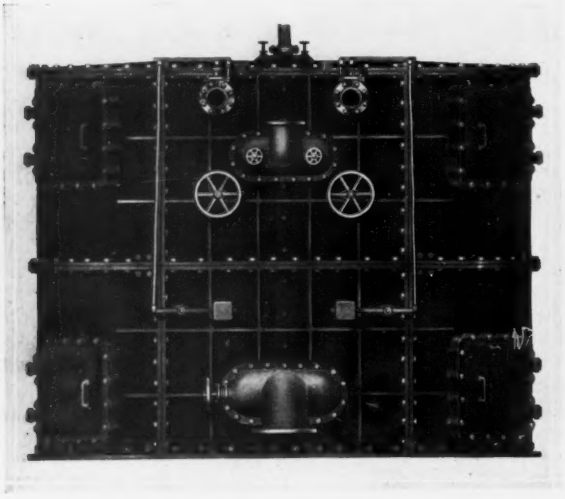


FIG. 1. FRONT ELEVATION OF THE ELLIOTT TWIN HEATER

trap, etc. The work of installation has been reduced to a minimum by simplifying the necessary piping, the complete valve equipment and connections forming part of the heater.

When in service, the steam enters the main inlet nozzle which communicates with 2 inlet nozzles on the 2 sides of the dividing wall, where it comes in contact with the combined valves and separators, which precipitate the entrainment into the overflow boxes below.

If the heater is carrying back pressure, a trap is attached to the 2 overflow boxes and serves as a means for sealing the openings; at the same time it automatically discharges the condensation water. The trap also relieves the heater of surplus water should the height of the normal water level be exceeded.

Cold water enters the float controlled inlet valves, is distributed to the pans, and heated by direct contact with the exhaust steam as it passes from the

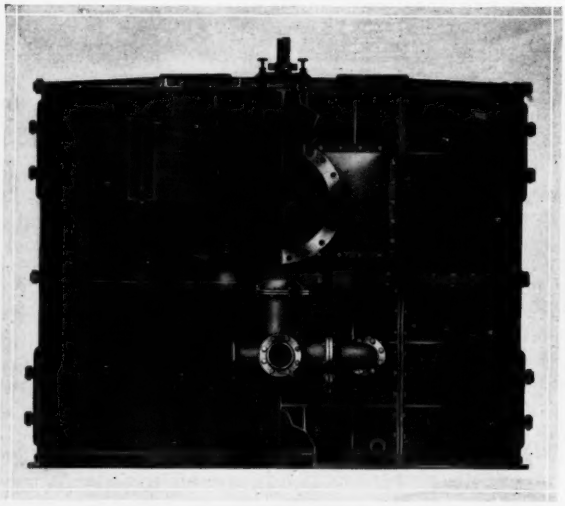


FIG. 2. REAR ELEVATION, HALF SECTION OF THE TWIN HEATER

spray box to the distributing pan over the cascade and settling pans until it falls into the water chamber beneath. The hot water outlet is hooded and vented

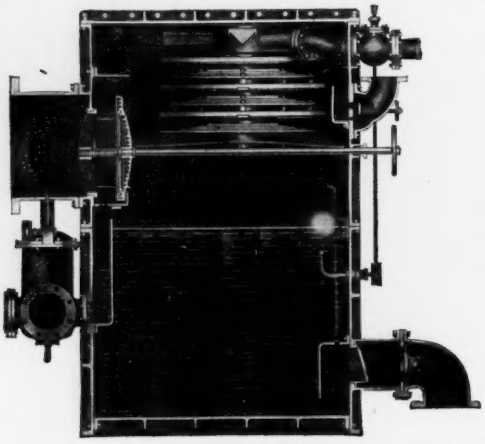


FIG. 3. TRANSVERSE SECTION OF TWIN HEATER, SHOWING GENERAL ARRANGEMENT OF COMBINED CUTOUT VALVE AND OIL SEPARATOR, TRAP, OVERFLOW BOX, PANS, PUMP SECTION, ETC.

to the steam space extending above the top of the water level, the 2 outlets communicating with the main pump suction being controlled by check valves.

The heater is vented through the outlet on top, which communicates with both sides through openings controlled by valves.

The cutout valves are essential to the twin heater, in order to control the steam supply to both sides. Each valve consists of a bronze seat inserted in each inlet nozzle, having a bar extending across it which is threaded and engages the thread on the end of the valve stem which passes through a combined separator plate and valve disc, terminating in a valve wheel on the opposite side of the heater.

The combined separator plate is mounted to slide on studs, and has its front face concave and provided with grooves to catch and retain the oil and entrainment, impinging upon it with the entering steam. Surrounding the combined valve and separator, is an annular plate used to change the direction of the steam to cause it to flow toward the center of the heater.

The pump suction connection on the twin heater extends downward and is provided with a flange. Each side has an independent outlet communicating with a

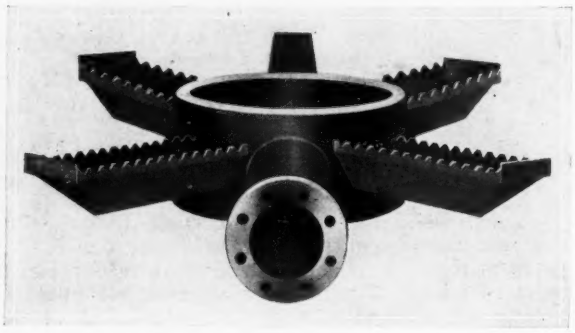


FIG. 4. SPRAY BOX, SHOWING FLANGED INLET AND SUPPORT

common chamber adjacent to the main outlet to the pump suction. The outlets to both sides of the heater are protected by check valves, opening in an outward direction. The check valves perform the double service of preventing water from backing up into the heater when one side is out of service, and assisting in maintaining the same water level on both sides. If for any reason the water in one side of the heater should drop to a lower point than in the other, as is the case when one side is shut off for cleaning purposes, or when the blowoff valve is open, the higher head on the opposite side immediately causes the check valve on the lower side to close and remain closed until the water level on that side becomes normal.

There is also a sliding gate valve for completely isolating either side of the heater. This is so arranged as to cut off either side, making it absolutely independent of the other side, but is rarely used except in emergencies, as in the event of the check valve sticking or leaking.

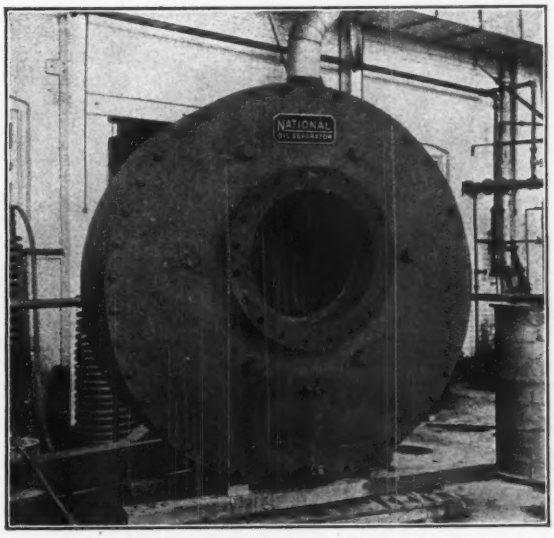
Entrainment coming from the separator, as well as the overflow from the heater, is controlled by a float actuated trap connected to the outlets of the 2 overflow boxes located below the steam inlet. The connection between the overflow outlets and the trap is provided with check valves opening in a direction toward the trap. The trap is vented to the main inlet nozzle. When one side of the heater is open for service, the water within the trap chamber, or the water in the trap chamber combined with the back pressure on the active side, causes the check valve on the dormant side to remain tight. When it is desired to cut out one side of the heater with this arrangement, all that is necessary is to close the cutout valve.

If the heater is used on a heating system, returns are admitted through a single inlet nozzle mounted on the front wall. This nozzle has 2 valve controlled branches, communicating one to each side of the heater. The return inlets are water sealed and placed in the steam chamber, and the water dropping to the water level has its temperature raised should its incoming temperature be less than that of the steam in the heater.

The twin heater is particularly well adapted to central stations, office buildings, or power plants in continuous operation. There are always light load periods during the day, when one side can be cut out and cleaned, at the same time maintaining the temperature of the water going to the boilers.

NATIONAL OIL SEPARATOR FOR PORTSMOUTH NAVY YARD

THE accompanying illustration shows the large National oil separator recently furnished the Portsmouth Navy Yard. The photograph was made, just before shipment, in the shops of the builders, the National Pipe Bending Co., New Haven, Conn. This 24-in. separator has an outside diameter of 72 in. and weights 5300 lb. The internal area of this relatively new type of separator is unusually large so that the incoming steam expands and is so reduced in velocity that anything heavier than steam, such as oil or water, will fall out of the current by gravity. Any minute particles which may be carried in the current are thrown out when the current impinges on the main baffle which extends the full width of the separator and has vertical ribs.



LARGE OIL SEPARATOR FOR PORTSMOUTH NAVY YARD

The distinctive feature lies in the arrangement of ports and the independent baffles which so oppose the current of steam that there is no chance of oil being carried by into the outlet chamber. There can be no drop in pressure because there is no friction; the area of the outlets is 1½ times greater than that of the inlet.

The National oil separator is made in 2 parts, which allows for inspection and testing of baffle plates and insures a perfect casting.

WHEELER TURBO AIR PUMP

IN general appearance this air pump, which is manufactured by the Wheeler Condenser & Engineering Co., of Carteret, New Jersey, is similar to a centrifugal pump, and is driven at high speed by either a motor or a turbine. The pump shown diagrammatically in Fig. 1 consists of a small high-speed impeller with a large number of ports or nozzles on its periphery; outside this is a ring of tapered compression channels, and beyond, a discharge casing or volute (not shown). Hurling or throwing water enters the

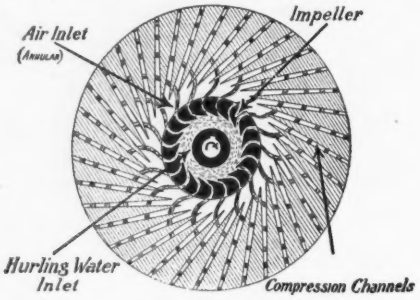


FIG. 1. DIAGRAM SECTION SHOWING PRINCIPLE OF WHEELER TURBO AIR PUMP

eye of the impeller as in an ordinary side suction centrifugal pump and is impelled at high velocity through the ports in the impeller, which rotates in the opposite direction to the impeller of a centrifugal pump.

Air is admitted around the entire periphery of the impeller and is drawn into the entrance of the tapered compressor channels by the aspirating effect of the rapidly discharged water. As the slugs of water move outward they become elongated owing to the tapering of the channels, the space between slugs decreases and the air entrapped is compressed. The mixture of air

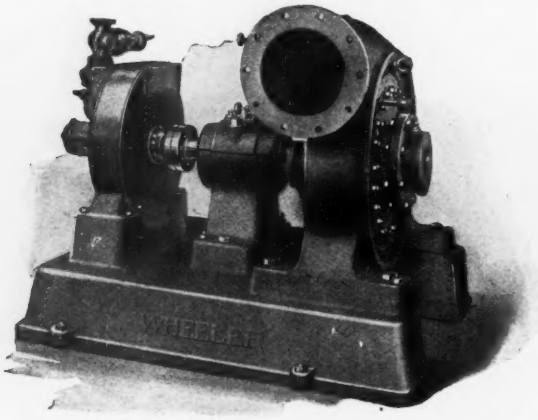


FIG. 2. COMBINED CONDENSATE AND AIR PUMP

and water is discharged from the tapered channels into a volute casing, somewhat similar to that of a centrifugal pump, and is finally discharged through a suitable outlet nozzle.

These pumps are used for both jet and surface condensers. For the latter service, the turbo air pump is frequently combined within a common casing with the condensate pump, thus securing many obvious advantages. Figures 2 and 3 illustrate a combined pump of this type. Air and condensate are withdrawn from the surface condenser through a common pipe, owing to a common suction nozzle on the pump. Just within the nozzle is a partition which separates air and water, the latter flowing by gravity to the eye of the condensate impeller, which is of the ordinary centrifugal pump design, while the air and gases flow over the partition wall, direct to the annular air inlet surrounding the hurling water impeller of the air pump.

Within the single casting 2 volutes are formed, one for condensate, the other for hurling water and air, so that as far as operation is concerned, the 2 pumps are entirely independent.

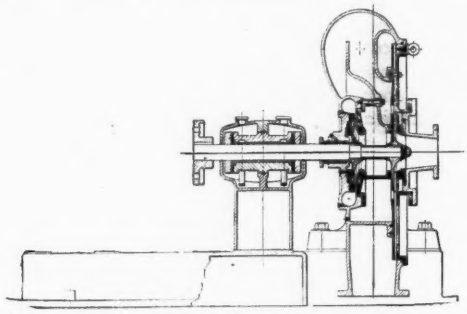


FIG. 3. SECTION OF COMBINED PUMP OF FIG. 2

A distinct saving results, however, from the fact that the amount of piping between condenser and pump is reduced and simplified, the space ordinarily required by the 2 pumps for air and condensate is greatly reduced, and both pumps are driven on a single shaft by a common turbine, thus saving power.

Pumps of this type are found to maintain a vacuum of 98 to 99 per cent of theoretical, that is within a small fraction of an inch of the vacuum corresponding to the temperature of the hurling or throwing water, not only for low temperatures of water from 32 to 60 deg., but also for hurling water at temperatures of 70, 80, 90 deg. and higher as encountered in the summer months and in cooling tower work.

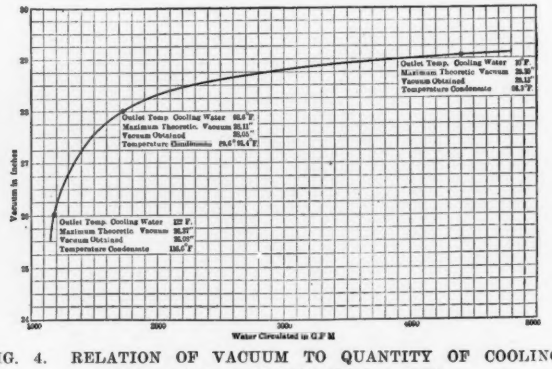

Vacuum in Inches
Water Circulated in G.P.M.
Outlet Temp. Cooling Water 87 F.
Maximum Theoretic Vacuum 28.39"
Vacuum Obtained 28.15"
Temperature Condensate 98.7 F.
Outlet Temp. Cooling Water 85.8 F.
Maximum Theoretic Vacuum 28.11"
Vacuum Obtained 28.03"
Temperature Condensate 98.6 F.
Outlet Temp. Cooling Water 32 F.
Maximum Theoretic Vacuum 28.87"
Vacuum Obtained 28.82"
Temperature Condensate 126.5 F.

FIG. 4. RELATION OF VACUUM TO QUANTITY OF COOLING WATER, INLET TEMPERATURE 50 DEG.

Large free air handling capacity follows from the fact that there is a positive aspirating effect at the entrance of the numerous tapered compression channels, into which air must be drawn so long as the impeller rotates and discharges water. Hence, increase in load or the cubic feet of free air handled, does not cause failure of the pump, but simply a fall in vacuum not unlike the characteristic of a displacement type pump.

The operating importance of this point is that, if the air leakage on a condenser be increased at any

time, due, for instance, to failure of the turbine seals, the pump will not refuse to handle the added load with a resulting break in vacuum, but will maintain a moderate vacuum until the air leakage trouble has been corrected.

Figure 4 shows the relation between vacuum maintained by a surface condenser equipped with a turbo air pump of this type and the amount of condenser circulating water in gallons a minute. The vacuum maintained and outlet circulating water temperatures are given for a number of points on the chart; for instance, with 4400 gal. of water a minute, the vacuum maintained was 29.13 in. with outlet cooling water at 70 deg., which corresponds to a maximum theoretical vacuum of 29.30 in., which is 0.17 in. higher than the actual vacuum maintained. This chart is of additional interest in that it shows the increase in cooling water necessary to obtain very high vacuum. To maintain a 29.1-in. vacuum, it was necessary to use almost 4 times as much water as that for a 26-in. vacuum and 2.5 times as much water as for a 28-in. vacuum.

SOME REMARKABLE EXAMPLES OF SOOT

MANY of us are apt to think of soot as a carbon deposit, such as is given by a smoky lamp on the chimney. But, as a matter of fact, the deposits from boiler tubes have a higher value than lampblack, because they increase the depth of the stagnant gas film as explained below.

Many are under the impression that soot will collect on boiler tubes only when bituminous coal is burned; but the accompanying photographs prove that the deposits actually found on the outside of the tubes of water-tube boilers are composed of many substances besides lampblack and that practically all boilers, regardless of the kind of fuel burned, are subject to such deposits.

The photographs show that the deposits are made up largely of ash carried over by the draft and deposited upon the tubes, and then they also contain inorganic matter from the fuel, which volatilizes the fire and then condenses when coming in contact with the comparatively cold tubes.

The deposits first occur in the form of a dust or powder, which can be easily removed, but after a time it is a strong possibility that this powder will fuse together, giving some of the clinker formations shown. This clinkering is undoubtedly due to the fact that as the deposit becomes thicker, its insulating qualities are greater, and finally the accumulation gets to the point where the temperature is sufficient to fuse the material in the top layers.

The sooty deposits in modern boilers may be classified into molten slag which forms as stalactites on the lower row of tubes when set close to the fire, and this formation is not common and cannot be blown off;

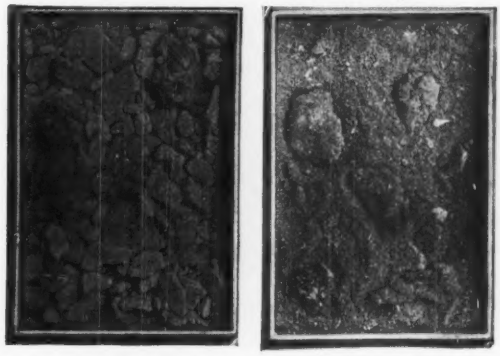


FIG. 3. DEPOSIT FROM THE FIRST PASS OF A HORIZONTAL WATER-TUBE BOILER, USING BUCKWHEAT COAL OF ABOUT 16 PER CENT ASH

FIG. 4. DEPOSIT FORMED ON TOP OF THE SCALE OF FIG. 3

clinker on top of the tube, found only in the first pass, as shown in Fig. 2; cemented mass of clinker, found only in lower rows of the first pass, as in Fig. 7; scaly deposit is due to moisture in the products of combustion, cementing the ash deposit which is frequently found throughout the boiler, as is shown in Fig. 3; loose soot, consisting of fine ashes, clay and other earthy matter.

Modern practices, such as the use of high draft and forcing of boilers at continuous overloads, are apt to increase the formation of sooting deposits. The same effect is produced by the burning of larger quantities of fuel per square foot of grate area. The samples of soot, photographs of which are here shown, are not selected ones, but represent the general run of deposits as found on boiler tubes. A harmful effect of such deposits, is that besides the direct heat insulating value, the porous character of the deposit forms a large number of small cells which retain the gases of combustion, and thus greatly increase the

FIG. 1. DEPOSIT FROM THE FIRST AND SECOND PASSES OF A MODEL WATER-TUBE BOILER, USING GARBAGE AS FUEL

FIG. 2. DEPOSIT FROM LOWER TUBES OF THE BOILER OF FIG. 1, SHOWING THE CLINKERING ACTION

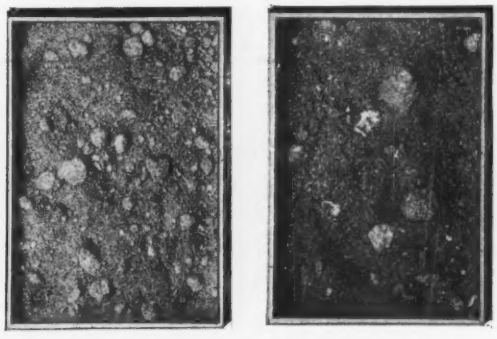


FIG. 5. DEPOSIT FROM THE TUBES OF A BOILER USING SHAWMUT COAL WHICH IS SOFT BUT GIVES A GRITTY ASH

FIG. 6. BLACK DEPOSIT FROM THE USE OF SHAWMUT COAL, MADE UP OF ASHES AND SOOT, WHICH WILL CLINKER WHEN SUBJECTED TO HEAT

depth of the stagnant gas film around the boiler tubes. The importance of this is shown by the fact that Prof. Dalby found that 90 per cent of the temperature head needed to force heat from the hot gases of combustion into the water on the inside of the tube, was used up in overcoming the resistance of this gas film. Mr. Hudson also found that the gas side of the plate form-

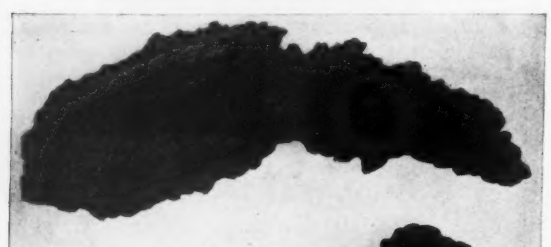


FIG. 7. SAMPLES OF THE LOWER ROW OF TUBES IN THE SAME PLANT, WHICH IS HARD, BRITTLE AND POROUS, AND LOOKS LIKE SLAG

ing the heating surface was never more than 36 deg. hotter than the water side, so that the loss of temperature from 1600 to 2000 deg. of the gases down to within 40 to 60 deg. of the temperature of the water, is due to loss through the gas film. By formation of porous clinker on the heating surfaces, the scrubbing action of high velocity gases in removing this film is nullified, and if the clinker gets thick enough to bridge over adjacent tubes, as shown by Fig. 7, it seriously interferes with the draft.

As mentioned above, the deposits at first are soft and easily blown off, so that the modern provision for frequent blowing off of boiler tubes becomes of im-

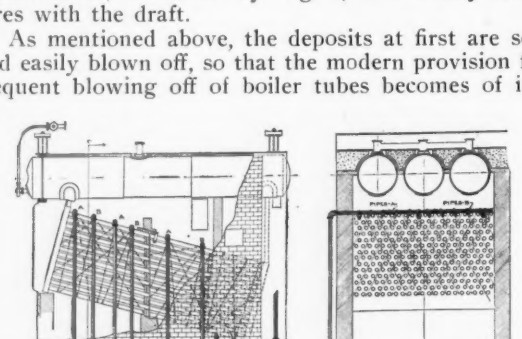


FIG. 8. VULCAN SOOT BLOWER, AS INSTALLED ON A B. & W. BOILER

portance. The old method of inserting a perforated steam pipe to blow off the soot is awkward, and as the pipe is hot, it is more or less dangerous, so that the job is apt to be shirked. It is also difficult, as no boiler 10 ft. wide can be cleaned by hand, unless alley spaces are left at the side, of more than usual width, thus wasting valuable ground space.

A convenient and satisfactory method of removing soot and ashes, is the soot cleaner shown in Fig. 8, which discharges steam through perforated pipes and nozzles, close to the tube surfaces, so that even the tubes in the center are effectively cleaned. Valves are provided, so that it is only necessary to open them one after the other to blow the soot all from the surfaces, and this permits of blowing frequently, the custom being to blow them once every watch. This kind of blowing, also, does not reduce furnace efficiency, as no doors have to be opened, permitting the inflow of air, and the saving is indicated by the fact that for a month's use at one plant, the average leaving temperature of the flow gases was 517.8 deg., and the carbon dioxide average 11.2 per cent, without blower in use.

A similar set of readings with a Vulcan soot cleaner in operation, and the boiler blown every day, showed an average leaving temperature of 391.83, and a carbon dioxide average of 11.8 per cent. This stack averaging 120 deg. lower, corresponds roughly to a saving of 6 per cent in fuel.

In a large railroad terminal, the use of the soot cleaners showed a saving of 7 per cent in the fuel bill, and other tests have shown from 7½ to 9 per cent saving. The samples of soot from which the photographs were taken were secured by the engineers of the New York office of the Vulcan Soot Cleaner Co., who have also furnished the data of this article.

BOOKS AND CATALOGS

THIRTY-FIFTH EDITION OF STEAM, 1913, by The Babcock & Wilcox Co., 7¼ by 10¼ in., 335 pages, illustrated.

This book, which is a treatise on the generation and use of steam, is much larger than and a complete revision of former editions containing information which has but recently come to light. Starting with the early history of the generation and use of steam, followed by a history of water-tube boilers, requirements of steam boilers and the evolution of the Babcock & Wilcox boiler, the authors lay the proper foundation for the study of steam generation and its use from the standpoint of the man in the power plant. Then follow separate chapters on such topics as Heat and Its Measurement, The Theory of Steam Making, Properties of Water, Boiler Feed Water, Steam, Moisture in Steam, Superheated Steam, Properties of Air, Combustion, Classification of Fuels, etc.; in fact, everything which has to do with the generation of steam comes in for its share of treatment in a clear and concise manner which will appeal to all connected with the power plant.

"THE LAGONDA Roller Bearing Cleaner" is the title of Bulletin X, recently issued by the Lagonda Mfg. Co., of Springfield, Ohio, describing a new type of water driven boiler tube cleaner. In this cleaner the old type thrust bearings have been replaced by tapered roller bearings. It is pointed out in the bulletin that bearings of this type are admirably adapted for use on automobiles, where the bearings are subjected to severe thrusts. It is claimed that this cleaner develops greater power and at a higher efficiency than any other type of cleaner so far developed.

Bulletin Y, entitled "The Lagonda Air Cleaner," issued by the same company, describes a new type of air driven boiler tube cleaners with several novel features. One is the use of an adjustable air admission port in the corner, by means of which the size of the air opening can be adjusted to meet the air pressure conditions in any given plant, thus insuring that the cleaner develops the maximum amount of power for the quantity of air consumed under all conditions. To dismantle the cleaner it is only necessary to loosen

These tables contain data especially worked out or carefully selected that are vital to your work. You may need them at any time and should preserve them. A handy, serviceable, loose-leaf cover in which these tables fit exactly, will be sent you for 30 cents in currency or stamps

SPECIFICATIONS—STANDARD RETURN TUBULAR BOILERS
125 POUNDS WORKING PRESSURE.

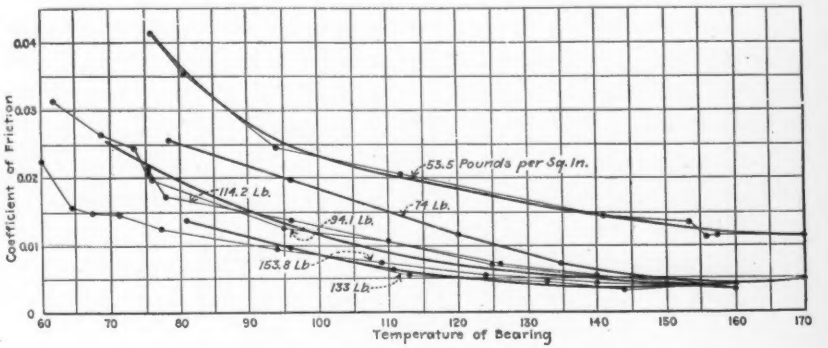
Longitudinal Seams, All Butt Jointed—Triple Riveted.

Rated HP. of Boiler	45	50	60	70	80	90	100
Dia. of Boiler ... in.	48	48	54	54	60	60	66
Length of Tubes ... ft.	12	14	14	16	16	18	16
No. of 3-in. tubes	46	46					
No. of 3½-in. tubes	34	34	44	44	54	54	66
No. of 4-in. tubes	28	28	36	36	44	44	54
Tube Htg. Surf., 3-in tubes ... sq. ft.	433.5	505.8					
Tube Htg. Surf., 3½-in. tubes ... sq. ft.	373.9	436.2	564.4	645.1	791.7	890.6	967.6
Tube Htg. Surf., 4-in. tubes ... sq. ft.	351.9	410.5	527.8	603.2	737.2	829.4	904.8
Shell Htg. Surf ... sq. ft.	100.5	117.3	131.9	150.8	167.6	188.5	184.3
Thickness, Shell ... in.	7/16	7/16	1/2	9/16	5/8	5/8	11/16
Thickness, Heads ... in.	7/16	7/16	7/16	7/16	7/16	7/16	7/16
Width of Grates ... in.	48	48	54	54	60	60	66
Length of Grates ... in.	42	48	48	54	54	60	54
Area of Grates ... sq. ft.	14.0	16.0	18.0	20.25	22.5	25.0	24.75
Dia. of Stack ... in.	24	24	26	26	28	28	30
Length of Stack ... ft.	40	50	50	60	60	60	60
Gage of Stack	16	16	16	16	14	14	14
Length of Guys ... ft.	240	300	300	600	600	600	600
Dia. of Guys ... in.	¼	¼	¼	¼	5/16	5/16	5/16

SIZE OF TRIMMINGS IN INCHES.

Steam Opening	3	3	3½	3½	4	5	5
Pop Safety Valve	2	2	2½	2½	2½	3	3
Water Gage Glass	¾	¾	¾	¾	¾	¾	¾
Water Column Con.	1	1	1¼	1¼	1¼	1¼	1¼
Gage Cocks	¾	¾	¾	¾	¾	¾	¾
Blow-off	1½	1½	2	2	2	2	2
Feed and Check Valves	1½	1½	1½	1½	1½	1½	1½
Steam Gage Dial	5	5	5	5	5	5	5

BEARING TEMPERATURE AND FRICTION FOR CUP GREASE. A. L. WESTCOTT



SPECIFICATIONS—STANDARD RETURN TUBULAR BOILERS
100 POUNDS WORKING PRESSURE.

Longitudinal Seams, All Butt Jointed—Double Riveted.

Rated HP. of Boiler	15	20	25	30	35	40	45	50	60	70
Dia. of Boiler ... in.	36	36	36	44	44	44	48	48	54	54
Length of Tubes ... ft.	8	10	12	10	12	14	12	14	14	16
No. of 3-in. tubes	24	24	24	36	36	36	46	46		
No. of 3½-in. tubes				28	28	28	34	34	44	44
No. of 4-in. tubes							28	28	36	36
Tube Htg. Surf., 3-in. tubes ... sq. ft.	150.8	188.5	226.2	282.7	339.2	395.8	433.5	505.8		
Tube Htg. Surf., 3½-in. tubes ... sq. ft.				256.6	307.9	359.2	373.9	436.2	564.4	645.1
Tube Htg. Surf., 4-in. tubes ... sq. ft.							351.9	410.5	527.8	603.2
Shell Htg. Surf ... sq. ft.	50.3	62.8	75.4	76.8	92.2	107.5	100.5	117.3	131.9	150.8
Thickness, Shell ... in.	1/4	1/4	1/4	1/4	1/4	1/4	5/16	5/16	5/16	5/16
Thickness, Heads ... in.	1/4	1/4	1/4	1/4	1/4	1/4	5/16	5/16	5/16	5/16
Width of Grates ... in.	36	36	36	42	42	42	48	48	54	54
Length of Grates ... in.	36	36	42	36	42	48	42	48	48	54
Area of Grates ... sq. ft.	9.0	9.0	10.5	10.5	12.25	14.0	14.0	16.0	18.0	20.25
Dia. of Stack ... in.	16	16	16	22	22	22	24	24	26	26
Length of Stack ... ft.	30	35	40	35	40	50	40	50	50	60
Gage of Stack	16	16	16	16	16	16	16	16	16	16
Length of Guys ... ft.	180	210	240	210	240	300	240	300	300	600
Dia. of Guys ... in.	5/16	5/16	5/16	¼	¼	¼	¼	¼	¼	¼

SIZE OF TRIMMINGS IN INCHES.

Steam Opening	2½	2½	2½	2½	3	3	3	3	3½	3½
Pop Safety Valve	1½	1½	1½	1½	1½	2	2	2	2½	2½
Water Gage Glass	¾	¾	¾	¾	¾	¾	¾	¾	¾	¾
Water Column Con.	1	1	1	1	1	1	1	1	1¼	1¼
Gage Cocks	¾	¾	¾	¾	¾	¾	¾	¾	¾	¾
Blow-off	1½	1½	1½	1½	1½	1½	1½	1½	2	2
Feed and Check Valves	1	1	1	1¼	1¼	1¼	1¼	1¼	1¼	1¼
Steam Gage Dial	5	5	5	5	5	5	5	5	5	5

SPECIFICATIONS—STANDARD RETURN TUBULAR BOILERS
150 POUNDS WORKING PRESSURE.

Longitudinal Seams, All Butt Jointed—Quadruple Riveted.

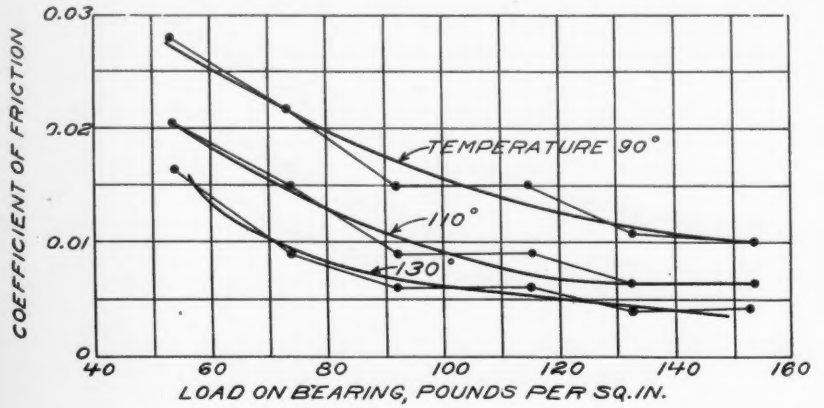
Rated HP. of Boiler	45	50	60	70	80	90	100
Dia. of Boiler ... in.	48	48	54	54	60	60	66
Length of Tubes ... ft.	12	14	14	16	16	18	16
No. of 3-in. tubes	46	46					
No. of 3½-in. tubes	34	34	44	44	54	54	66
No. of 4-in. tubes	28	28	36	36	44	44	54
Tube Htg. Surf., 3-in. tubes ... sq. ft.	433.5	505.8					
Tube Htg. Surf., 3½-in. tubes ... sq. ft.	373.9	436.2	564.4	645.1	791.7	890.6	967.6
Tube Htg. Surf., 4-in. tubes ... sq. ft.	351.9	410.5	527.8	603.2	737.2	829.4	904.8
Shell Htg. Surf ... sq. ft.	100.5	117.3	131.9	150.8	167.6	188.5	184.3
Thickness, Shell ... in.	11/16	11/16	3/4	13/16	7/8	7/8	15/16
Thickness, Heads ... in.	7/16	7/16	7/16	1/2	1/2	1/2	1/2
Width of Grates ... in.	48	48	54	54	60	60	66
Length of Grates ... in.	42	48	48	54	54	60	54
Area of Grates ... sq. ft.	14.0	16.0	18.0	20.25	22.5	25.0	24.75
Dia. of Stack ... in.	24	24	26	26	28	28	30
Length of Stack ... ft.	40	50	50	60	60	60	60
Gage of Stack	16	16	16	16	14	14	14
Length of Guys ... ft.	240	300	300	600	600	600	600
Dia. of Guys ... in	¼	¼	¼	¼	5/16	5/16	5/16

SIZE OF TRIMMINGS IN INCHES.

Steam Opening	3	3	3½	3½	4	5	5
Pop Safety Valve	2	2	2½	2½	2½	3	3
Water Gage Glass	¾	¾	¾	¾	¾	¾	¾
Water Column Con.	1	1	1¼	1¼	1¼	1¼	1¼
Gage Cocks	¾	¾	¾	¾	¾	¾	¾
Blow-off	1½	1½	2	2	2	2	2
Feed and Check Valves	1½	1½	1½	1½	1½	1½	1½
Steam Gage Dial	5	5	5	5	5	5	5

These tables contain data especially worked out or carefully selected that are vital to your work. You may need them at any time and should preserve them. A handy, serviceable, loose-leaf cover in which these tables fit exactly, will be sent you for 30 cents in currency or stamps.

BEARING PRESSURE AND FRICTION FOR CUP GREASE. A. L. WESTCOTT



SPECIFICATIONS—STANDARD RETURN TUBULAR BOILERS
125 POUNDS WORKING PRESSURE.

Longitudinal Seams, All Butt Jointed—Triple Riveted.

Rated HP. of Boiler	110	125	150	165	180	200
Dia. of Boiler...in.	66	72	72	72	78	78
Length of Tubes...ft.	18	16	18	20	18	20
No. of 3-in. tubes						
No. of 3 ½-in. tubes	66	86	86	86	110	110
No. of 4-in. tubes	54	70	70	70	88	88
Tube Htg. Surf., 3 in. tubes...sq. ft.						
Tube Htg. Surf., 3 ½-in. tubes...sq. ft.	1088.6	1260.8	1418.4	1576.0	1814.0	2015.9
Tube Htg. Surf., 4-in. tubes...sq. ft.	1017.9	1172.9	1319.5	1466.0	1658.8	1843.1
Shell Htg. Surf...sq. ft.	207.4	201.1	226.2	251.3	245.2	272.4
Thickness, Shell...in.	11/32	11/32	11/32	11/32	11/32	11/32
Thickness, Heads...in.	1/2	1/2	1/2	1/2	9/16	9/16
Width of Grates...in.	66	72	72	72	78	78
Length of Grates...in.	60	54	60	66	60	66
Area of Grates...sq. ft.	27.5	27.0	30.0	33.0	32.5	35.75
Dia. of Stack...in.	30	34	34	34	38	38
Length of Stack...ft.	60	60	60	70	60	70
Gage of Stack	14	14	14	14	12	12
Length of Guys...ft.	600	600	600	700	600	700
Dia. of Guys...in.	5/16	5/16	5/16	5/16	3/8	3/8

SIZE OF TRIMMINGS IN INCHES.

Steam Opening	5	5	6	6	6	6
Pop Safety Valve	3	4	4	4	4	4
Water Gage Glass	3/4	3/4	3/4	3/4	3/4	3/4
Water Column Con.	1½	1½	1½	1½	1½	1½
Gage Cocks	3/4	3/4	3/4	3/4	3/4	3/4
Blow-off	2	2	2½	2½	2½	2½
Feed and Check Valves	1½	1½	1½	1½	2	2
Steam Gage Dial	5	5	5	5	5	5

SPECIFICATIONS—STANDARD RETURN TUBULAR BOILERS
150 POUNDS WORKING PRESSURE.

Longitudinal Seams, All Butt Jointed—Quadruple Riveted.

Rated HP. of Boiler	110	125	150	165	180	200
Dia. of Boiler...in.	66	72	72	72	78	78
Length of Tubes...ft.	18	16	18	20	18	20
No. of 3-in. tubes						
No. of 3 ½-in. tubes	66	86	86	86	110	110
No. of 4-in. tubes	54	70	70	70	88	88
Tube Htg. Surf., 3 in. tubes...sq. ft.						
Tube Htg. Surf., 3 ½-in. tubes...sq. ft.	1088.6	1260.8	1418.4	1576.0	1814.0	2015.9
Tube Htg. Surf., 4-in. tubes...sq. ft.	1017.9	1172.9	1319.5	1466.0	1658.8	1843.1
Shell Htg. Surf...sq. ft.	207.4	201.1	226.2	251.3	245.2	272.4
Thickness, Shell...in.	11/32	11/32	11/32	11/32	11/32	11/32
Thickness, Heads...in.	1/2	1/2	1/2	1/2	9/16	9/16
Width of Grates...in.	66	72	72	72	78	78
Length of Grates...in.	60	54	60	66	60	66
Area of Grates...sq. ft.	27.5	27.0	30.0	33.0	32.5	35.75
Dia. of Stack...in.	30	34	34	34	38	38
Length of Stack...ft.	60	60	60	60	60	70
Gage of Stack	14	14	14	14	12	12
Length of Guys...ft.	600	600	600	700	600	700
Dia. of Guys...in.	5/16	5/16	5/16	5/16	3/8	3/8

SIZE OF TRIMMINGS IN INCHES.

Steam Opening	5	5	6	6	6	6
Pop Safety Valve	3	4	4	4	4	4
Water Gage Glass	3/4	3/4	3/4	3/4	3/4	3/4
Water Column Con.	1½	1½	1½	1½	1½	1½
Gage Cocks	3/4	3/4	3/4	3/4	3/4	3/4
Blow-off	2	2	2½	2½	2½	2½
Feed and Check Valves	1½	1½	1½	1½	2	2
Steam Gage Dial	5	5	5	5	5	5

SPECIFICATIONS—STANDARD RETURN TUBULAR BOILERS
100 POUNDS WORKING PRESSURE.

Longitudinal Seams, All Butt Jointed—Double Riveted.

Rated HP. of Boiler	80	90	100	110	125	150	165	180	200
Dia. of Boiler...in.	60	60	66	66	72	72	72	78	78
Length of Tubes...ft.	16	18	16	18	16	18	20	18	20
No. of 3-in. tubes									
No. of 3 ½-in. tubes	54	54	66	66	86	86	86	110	110
No. of 4-in. tubes	44	44	54	54	70	70	70	88	88
Tube Htg. Surf., 3 in. tubes...sq. ft.									
Tube Htg. Surf., 3 ½-in. tubes...sq. ft.	791.7	890.6	967.6	1088.6	1260.8	1418.4	1576.0	1814.0	2015.9
Tube Htg. Surf., 4-in. tubes...sq. ft.	737.2	829.4	904.8	1017.9	1172.9	1319.5	1466.0	1658.8	1843.1
Shell Htg. Surf...sq. ft.	167.6	188.5	184.3	207.4	201.1	226.2	251.3	245.2	272.4
Thickness, Shell...in.	5/16	5/16	11/32	11/32	3/8	3/8	3/8	13/32	13/32
Thickness, Heads...in.	7/16	7/16	7/16	7/16	1/2	1/2	1/2	1/2	1/2
Width of Grates...in.	60	60	66	66	72	72	72	78	78
Length of Grates...in.	54	60	54	60	54	60	66	60	66
Area of Grates...sq. ft.	22.5	25.0	24.75	27.5	27.0	30.0	33.0	32.5	35.75
Dia. of Stack...in.	28	28	30	30	34	34	34	38	38
Length of Stack...ft.	60	60	60	60	60	60	70	60	70
Gage of Stack	14	14	14	14	14	14	14	12	12
Length of Guys...ft.	600	600	600	600	600	600	700	600	700
Dia. of Guys...in.	5/16	5/16	5/16	5/16	5/16	5/16	5/16	3/8	3/8

SIZE OF TRIMMINGS IN INCHES.

Steam Opening	4	5	5	5	5	6	6	6	6
Pop Safety Valve	2½	3	3	3	4	4	4	4	4
Water Gage Glass	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4
Water Column Con.	1½	1½	1½	1½	1½	1½	1½	1½	1½
Gage Cocks	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4
Blow-off	2	2	2	2	2	2½	2½	2½	2½
Feed and Check Valves	1¼	1¼	1½	1½	1½	1½	1½	2	2
Steam Gage Dial	5	5	5	5	5	5	5	5	5

a hexagonal retaining ring at the front which renders all of the interior parts of the cleaner readily accessible without the use of any tools.

Copies of these bulletins may be had by addressing the company as above.

"A PIPE WITHOUT a Parallel Which Will Last a Lifetime," is the title of a circular lately issued by A. M. Byers Co., 235 Water St., Pittsburgh, Pa.

GAS ENGINES for any fuel gas, such as producer, blast furnace, coke oven or natural gas, in sizes from 350 b.hp. upwards, are described in Bulletin I from Mesta Machine Co., Pittsburgh, Pa.

N. T. C. BULLETIN NO. 16, on "National" Stationary and Marine Boiler Tubes, deals with the tests to which "National" boiler tubes are subjected. A copy will be mailed on request to National Tube Co., Frick Bldg., Pittsburgh, Pa.

AN UNUSUAL BOOKLET, whose cover design is illustrated herewith, has been received from Ingersoll-Rand Co., 11 Broadway, N. Y. Analogous to the silkworm's development, the story of ore is presented in attractive form from its entering the cocoon, or steel mill, to its final abode—the Butterfly rock drill, stope drill, plug drill or jackhamer.

CIRCULAR No. 39A from Crane Co., Chicago, describes and illustrates Crane combination back pressure and exhaust relief valves for condensing and non-condensing engines, and automatic-atmospheric exhaust relief valves for condensing engines only.

THE TALE OF THE THREE, giving the autobiographies of 3 drops of lubricant, begins in the October number of the Bearing, published by Albany Lubricating Co., 708 Washington St., New York.

COMPARATIVE TESTS, by Robert Thurston Kent, of new leather belts with and without Cling-Surface treatment are presented in a pamphlet recently issued by Cling-Surface Co., Buffalo, N. Y.

FROM THE ENGINEERING DEPARTMENT of Edge Moor Iron Co., Edge Moor, Del., has been received a copy of Bulletin 51, which is the report of tests of a steam boiler at high rates of evaporation, made at the Westport Generating Station of the Consolidated Gas, Electric Light & Power Co., Westport, Baltimore. This boiler is equipped with a Foster superheater and Taylor underfeed stoker. The rating is based upon 10 sq. ft. of heating surface per horsepower, and is 736.1 hp. Among the notable results of this test are, that the boiler was raised from banked fires to full rating in 2.3 min. and to 200 per cent rating in 7.8 min., while during the test the boiler was operated, during a period of the time, at 318 per cent of its rating. Those wishing a copy of this bulletin may secure it by writing to the company.

GOVERNMENT PURCHASING AGENTS, designing and operating engineers, and the fuel departments of industrial concerns, large dealers in coal, and persons interested in the distribution and character of the different coals in the United States will find valuable information in a report just issued by the United States Bureau of Mines as Bulletin 22, entitled "Analyses of coals in the United States, with descriptions of mine and field samples collected between July 1, 1904, and June 30, 1910." This report contains the analyses of 5000 samples of coal taken from 1500 coal mines and prospects situated in the various coal fields of the United States. Not only all of the important fields are represented, but practically all of the more important mining districts.

The purpose of the bureau in compiling and publishing this information is to present reliable information regarding the chemical composition and heating value of the coals. The samples of coals were collected by experienced men according to a definite and uniform system, and were analyzed under carefully controlled conditions, so that there might be no question as to the relative merits of the different coals so far as this can be determined by chemical analyses and determination of heating values.

TRADE NOTES

ALLIS-CHALMERS MANUFACTURING CO. announces the removal of the Sales and Engineering Offices to its Mining Machinery Department from Chicago to the Milwaukee Works at West Allis. The Chicago shops will also be removed to Milwaukee in the near future, to gain the advanages of concentrating all departments, both commercial and manufacturing, at one plant.

THE AMERICAN BLOWER CO., of Detroit, Michigan, have purchased the entire air washer interests, including patent rights, of the McCreery Engineering Co., formerly of Toledo, Ohio, and later of Detroit, Michigan, and the McCreery purifying, cooling and humidifying equipment will hereafter be exclusively manufactured and sold under the trademark Sirocco.

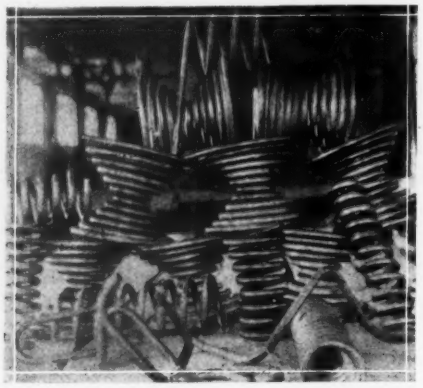
PURCHASE OF THE Clifford-Kapell Fan Co. by the Elliott Co., of Pittsburgh, is announced, and the plant at Jeannette, Pa., is now under course of remodeling. A new foundry, 300 by 150 ft., will be erected, and other buildings added to and built, to give an added floor space of about 2½ acres. In these works the manufacture of the Clifford fan for mine ventilation will be continued, and the manufacture of steam specialties made by the Elliott Co. will be carried on.

AT THE SPRINGFIELD convention, a valve was exhibited by the National Tube Co., which had been used on an outfit for testing Kewanee unions, and was, therefore, opened and closed every time a

union was tested. A walrus skin traveling bag was offered to the person making the nearest estimate of the total number of times that the valve had been operated, and was awarded to Wm. H. White, Superintendent of the Rochester Railway & Light Co., Rochester, N. Y., on an estimate of 2,648,500, the actual number of operations being 2,650,000.

FOR DRIVING ALL of the apparatus of the Mountain Ice Co., of Hoboken, N. J., electric power is to be furnished by a 300-kw. Terry steam turbine, direct-connected to a Crocker-Wheeler 375 kv. a. 60-cycle alternator running at 3600 r.p.m. This set will furnish current for the conveyor, the planer and the centrifugal pump for flushing away chips, also for driving the gang saws for cutting the ice, which is a novel application.

HEREWITH ARE SHOWN examples of coils from ¾-in. copper tube, No. 20 gage wound without the use of any filler in lengths up to 30 ft., the winding



of the double cone coils taking 20 sec. each. The process is patented by W. F. Smith of Baltimore, Md.

TWO TERRY TURBINE-DRIVEN forced draft blower sets, which are probably the largest in point of horsepower in the world, are about to be installed at the Rhode Island Co.'s power plant at Providence, R. I. Each unit consists of a 600-hp. Terry turbine, driving through a set of Falk reduction gears, an American Blower Co.'s No. 11 double-inlet Sirocco blower. Each unit has a capacity of 250,000 cu. ft. per min. and supplies draft to mechanical stokers. The turbines run at a speed of 3370 r.p.m., the blowers at 460 r.p.m., the speed reduction being 7.35 to 1.

BIDS FOR THE CONSTRUCTION of the proposed State Hospital power house and chimney stack at Warren, N. Y., were opened recently in the office of Superintendent Mitchell in the presence of the Board of Trustees. The contract for the power house was awarded to Henry Schenck and Co., of Erie, their bid of $35,590, being the lowest submitted. The contract for radial brick chimney for the power house was awarded to the Alphonse-Custodis Chimney Construction Co., on their bid of $6630. The power house will be located about 300 ft. north of the D. A. V. & P. station at North Warren and will be of brick, dimensions 60 by 40 ft. Under the contract it must be roofed over in 3 months. The chimney will be 185 ft. in height and must be completed in 42 working days.

PRACTICAL ENGINEER

Trade-mark reg. U. S. Pat. Office.

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Positions Wanted

POSITION WANTED—As mechanical draftsman and designer. College graduate, mechanical engineer. Completed two correspondence courses in drawing. Over year's practical drafting. A-1 references. Edward Skillman, Tribune, Kansas. 8-15-4

POSITION WANTED—Machinist, Engineer or Master Mechanic. Was erecting engines for Ingersoll-Rand Co. six years. First class man on all kinds of engines, air compressors and pumps. Can indicate and do all repairs; will go anywhere. Age. 35. Address Box 317, Practical Engineer.

POSITION WANTED—By first-class Steam Engineer. A. and D. current. Can give results and best of reference. Sober and steady, 14 yrs.' experience. Address Box 309, Practical Engineer, Chicago, Ill. 10-1-1

POSITION WANTED—Experienced salesman, 27, American, forceful and energetic, good personal appearance, desires connection with manufacturer as traveling representative in any territory. Wishes to learn line thoroughly so that success will be positive. Substantial references and surety bond. Address Salesman, No. 181 Mercer St., New York, N. Y. 10-15-1

POSITION WANTED—By a refrigeration engineer, who is a first class mechanic and understands thoroughly the economic operation, management, overhauling and repairing of a refrigeration plant, and has the energy and the ability to make good. Six years in present position. W. E. H. Box 318, Practical Engineer. 10-15-1

Wanted

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Sparks On The Advertising Pages

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Among the many excellent addresses delivered by Efficiency Experts, Editors, Educators, Advertising Men and Publishers, was "The New Force in Business" by Mr. R. R. Shuman, of the Shuman-Booth Co., Chicago. In speaking of the technical press, Mr. Shuman said:

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